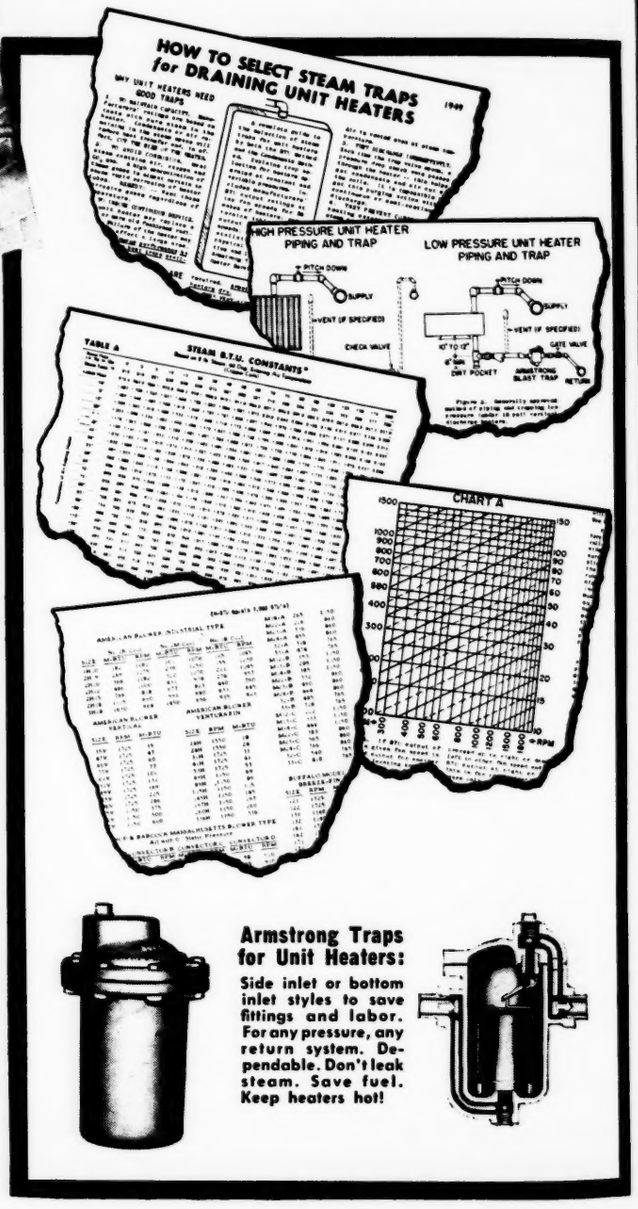
HEATING AND VENTILATING

HOW TO SELECT UNIT HEATER TRAPS

1. To obtain full BTU output
2. To insure continuous service
3. To avoid corrosion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THIS MONTH'S COVER

Restaurants and night clubs realize the importance of comfort air conditioning as a means of attracting customers during the summer. In designing cooling systems for such places, it is important to understand how to make heat gain calculations. See page 85. Photograph, Courtesy Billy Rose's Diamond Horseshoe, New York City.

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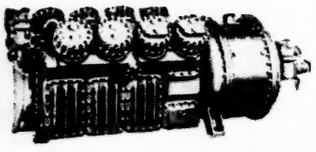

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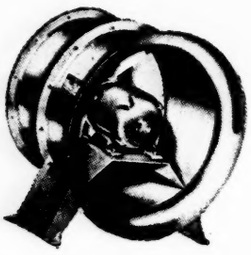

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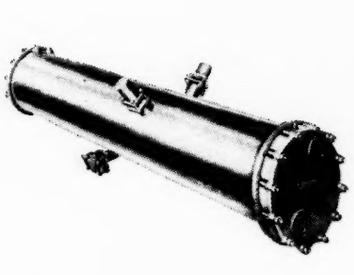

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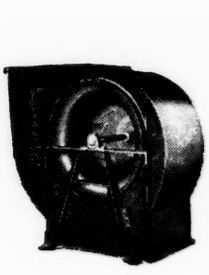

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Washington News

LORING F. OVERMAN

JULY was construction month in Washington for the building industry and its countless components—including heating, ventilating and air conditioning—with one multi-billion dollar program signed into law by the President, with two others given the go sign by Congressional committees, and with a fourth dumped into the Senate and House hoppers for early consideration.

Signed into law on July 15 was the Housing Act of 1949, authorizing Federal assistance in slum clearance, public housing, farm housing, and housing research.

Unanimously approved by the Senate Public Works Committee on July 22 was a proposal to authorize $100 million to assist states in advance planning of public works. This bill, while it has a long road to travel before final passage, is not considered controversial and has a chance of getting by before Congress adjourns. It is of considerable interest to readers of this column because of the type of construction contemplated. The bill would authorize the General Services Administration to make loans to states and their public works agencies to finance surveys, architectural studies, economic investigations, designs, and other preliminary actions for public works projects.

Also unanimously approved by the House Armed Services Committee was the $614 million military construction program. The committee will seek immediate House action on the bill, which authorizes military construction in and outside the United States, and includes more than 7,500 family housing units to be built on military posts at a cost of $150 million. This, too, would be permanent construction of a type intended to attract officer and enlisted personnel now said to be shying away from military service because of difficulty in securing suitable family accommodations adjacent to military posts.

Fourth of the measures of interest to the construction industries were identical bills introduced in the Senate by Senator Sparkman of Alabama, and in the House by Representative Spence of Kentucky. These bills, long and detailed in their provisions, concede that the Housing Act of 1949 takes care of the low income groups, so proposes that something be done for the moderate-income folks.

The measure (H.R. 5631 and S. 2246) propose, among other things:

To liberalize and extend the present system of mortgage insurance by the Federal Housing Administration, including Title I (modernization and low-cost housing in outlying areas); Title II (urban housing), and Section 608 (multi-family housing).

To remove certain limitations on the purchase of mortgages from private lending institutions by the Federal National Mortgage Association, a federal agency. The effect would be to facilitate the flow, by indirection, of federal money into private home building.

To liberalize home loan guarantees by the Veterans Administration and to authorize direct lending to veterans by the Veterans Administration if private investment funds at 4% interest are not forthcoming in adequate quantities.

To establish a new division in the Housing and Home Finance Agency, to be known as the Cooperative Housing Administration, with a fund of $1 billion for direct federal loans, at 3% for 60 years, to cover the full cost of cooperative housing projects. Rents, income limitations on tenants, design standards, etc., would be controlled by the federal government. Coupled with this is a liberalization of existing loan insurance for cooperatives through the FHA.

To provide for transfer to local public housing agencies, for use in the government low-rent housing program, of 120 listed permanent war housing projects built under the Lanham Act, upon formal request by local governments; for the sale of other permanent war housing units, and for the sale or demolition of temporary Lanham Act housing.

To provide $300 million for direct loans to educational institutions for construction of student and faculty housing.

There seems a likelihood that many of the provisions of this fourth measure will beat the adjournment deadline, for it continues many of the existing functions of the present Housing and Home Financing Agency. That this may have been anticipated by Congress is indicated by the fact that the Housing Act of 1949, Title II, provides for temporary extension (through August 31, 1949, pending further action by Congress) of FHA's Title I and Section 608 mortgage insurance operations; also for a $500 million increase in its Title II authorization. The extension under Title I relates to small loans for alteration, and improvement and new construction, and under Section 608 to rental housing. The increase in insurance authorization applies to all types of housing under Title II, owner-occupied, small and large scale rental and sale housing, including cooperative housing.

Housing Act Sounds Good

As explained by Walter Royal, public relations officer of the Housing and Home Finance Agency, the new Housing Act of 1949, signed by the President, sounds very logical and realistic. Recalling the bitter floor fights in the House and Senate, with "socialized housing" a frequently used term, Mr. Royal's explanation leaves the impression that the new program may not be so bad after all.

He stressed again and again that projects must be originated at the local level, that the local architect and engineer will presumably be sufficiently alert to be in on the planning stages and that local public hearings will determine the need for proposed projects and determine whether or not they are considered by private industry to be within its field. He also explained that, once a project is authorized and under way, all

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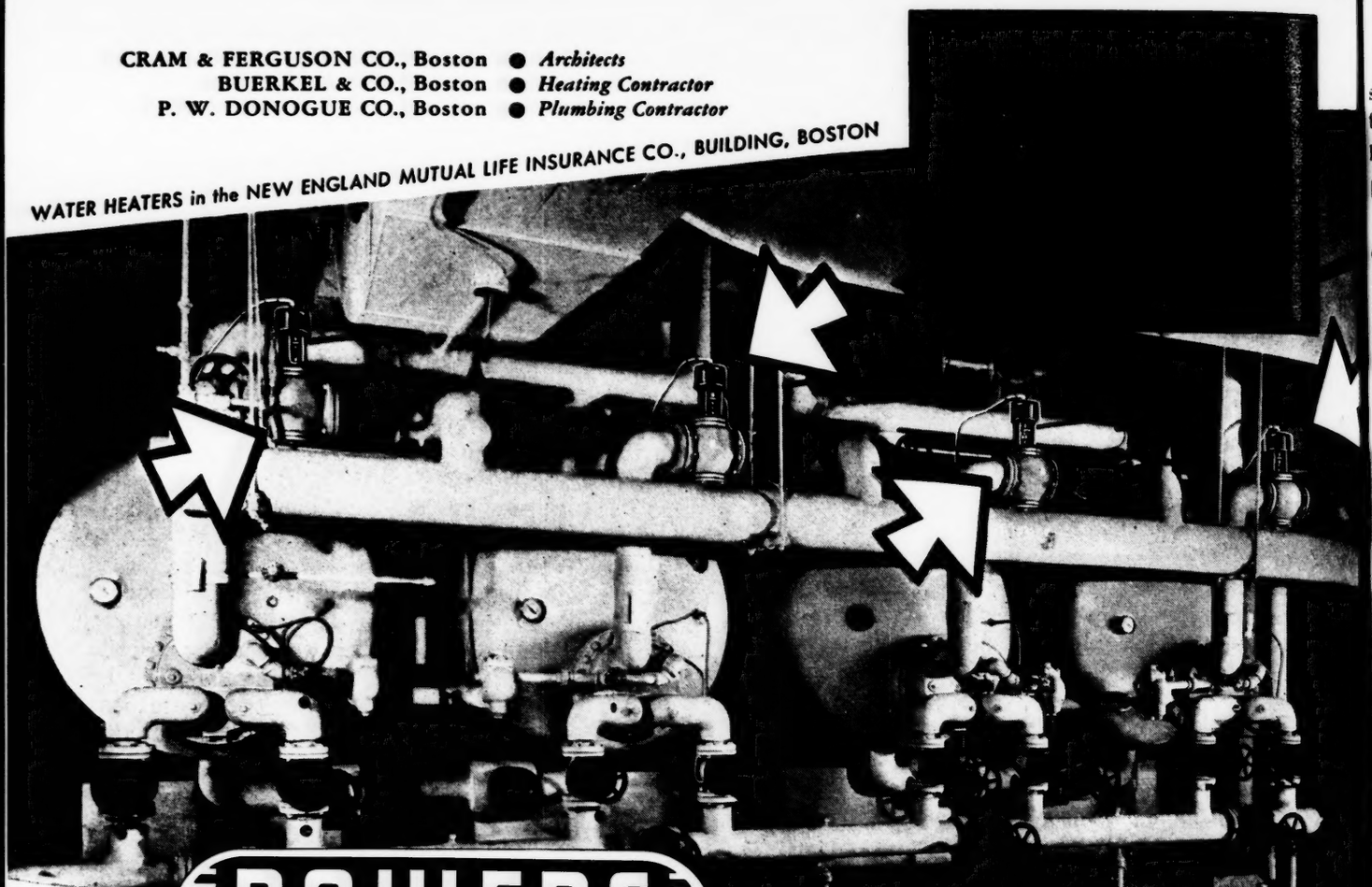

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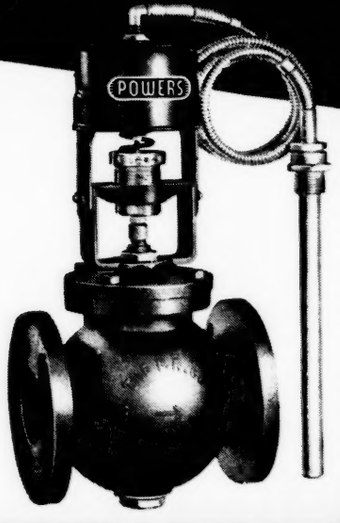

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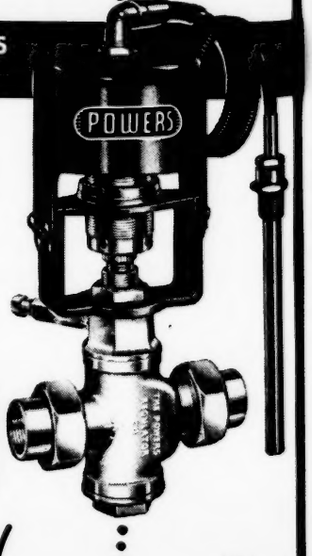

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Not New, Just Bigger

"There is nothing very new about the Housing Act of 1949," said Mr. Royal, "for it merely takes up in 1949 the old USHA program originated in 1937 and interrupted in 1940 by the defense program. There is one important difference—slum clearance is to be handled as a separate project by the Housing and Home Finance Agency, while low-rent housing is to be handled separately by the Public Housing Administration.

"Under the old USHA plan slum clearance and low-rent housing were handled together, with the requirement that areas cleared of substandard housing were customarily replaced, on the same site, with new low-rent housing. Current thinking is that the slum areas themselves—usually in locations not well suited to housing—can, in many cases, be cleared of below-standard housing and devoted to other and more logical purposes. The law provides, however, that provisions must be made for persons displaced by slum clearance programs, but not necessarily on the same site."

Slum Clearance Example

Under the new law, Title I—Slum Clearance and Community Development and Redevelopment — $1 billion is provided in loans over a five-year period to finance the planning of local projects, to acquire land through purchase or condemnation, and to clear the site for re-use. These loans would be repaid to the Federal Government from the proceeds of sale or lease of the cleared site. Also authorized is $500,000,000 in Federal capital grants over a five-year period to meet losses involved in connection with slum clearance operations.

Mr. Royal explained that these loans and capital grants are available only to local housing authorities, 472 of which have been authorized by law in 42 states. Action to make the other states eligible are contemplated soon in Oklahoma, Kansas, Iowa, Utah, Idaho and Wyoming.

"Let us assume," said Mr. Royal, "that the Omaha housing authority proposes to clear 10 square blocks near the packing houses. After hearings decide the proposal desirable, they can borrow from the Federal Government the funds to engineer the project and to determine costs and to make recommendations as to suitable uses for the cleared area. Step 2, having the plan, additional loans can be secured to acquire and clear the site. Step 3, the area is now cleared—ready for use by the city for a park or other public area, or to be sold or leased. Perhaps the cost of acquiring and clearing has been $1,000,000, but the returns from sale or lease only $500,000. The HHFA will then make a capital grant, or subsidy, covering two-thirds of the deficit.

"At this point the local construction industry comes into the picture, with a cleared site ready to be handled in the usual way, depending upon the use to be made of the property."

Title III of the law (Title II extended certain FHA mortgage provisions through August 31, 1949) amends the United States Housing Act of 1937 by authorizing Federal contributions and loans for local programs involving not more than 810,000 additional units of low cost public housing over a six-year period. The Public Housing Administration may authorize local authorities to commence construction of 135,000 units each year. The President, however, is authorized to accelerate the program to not more than 200,000 units per year or to retard the program to not less than 50,000 units per year, subject to the total authorization of 810,000 units; if it is determined that such action is in the public interest.

Public Housing Example

"For example," Mr. Royal continued, "let us suppose that in clearing its 10 square blocks, the Omaha authority found it would eliminate 1,000 living units. It would be necessary to replace substantially the same number, possibly on the same site, but more probably on some other site more suitable for residences. The Omaha authority will then make its needs known to the Public Housing Administration, which is authorized to make loans amounting to $1,500,000,000 for project planning purposes, to secure temporary notes sold by local authorities to investors, and to make permanent loans for any part of the permanent financing not covered by the sale of bonds to investors.

"Assuming that the 1,000 units are constructed at a cost making it necessary to charge $60 per unit to amortize, but that the low-income group for whom the project was intended can pay only $20 per month, there would be a deficit of $40 per month per unit. The Federal Government would pay approximately half of this monthly deficit as a capital grant subsidy for the entire amortization period—40 years. The amount of the Federal subsidy is limited to a percentage of the development cost of the project, and the total amount of subsidies may not exceed $308,000,000 per year."

In its explanation of the new program, the Housing and Home Finance Agency takes pains to point out that the new public housing program will not compete with, but will contribute to private building.

"The program," says an HHFA release, "will not compete with standard housing—just with slum housing. The system of income limits for occupants and the gap between public and private housing insure that public housing will serve only those families who cannot afford even the cheapest of the adequate housing provided by private enterprise. Since the tenants of public housing will be families who cannot afford standard housing, they are not a part of the market that private housing serves.

"All construction is done by private contractors who get the work by competitive bidding. This means business for private architects, engineers, materials manufacturers and suppliers. Since the projects will be financed in large part by private borrowings, this means business for bankers and investors. Other businesses in the community stand to gain because the tenants of public housing, paying rents commensurate with their incomes, will have enough money left over for such essentials as food and clothing."

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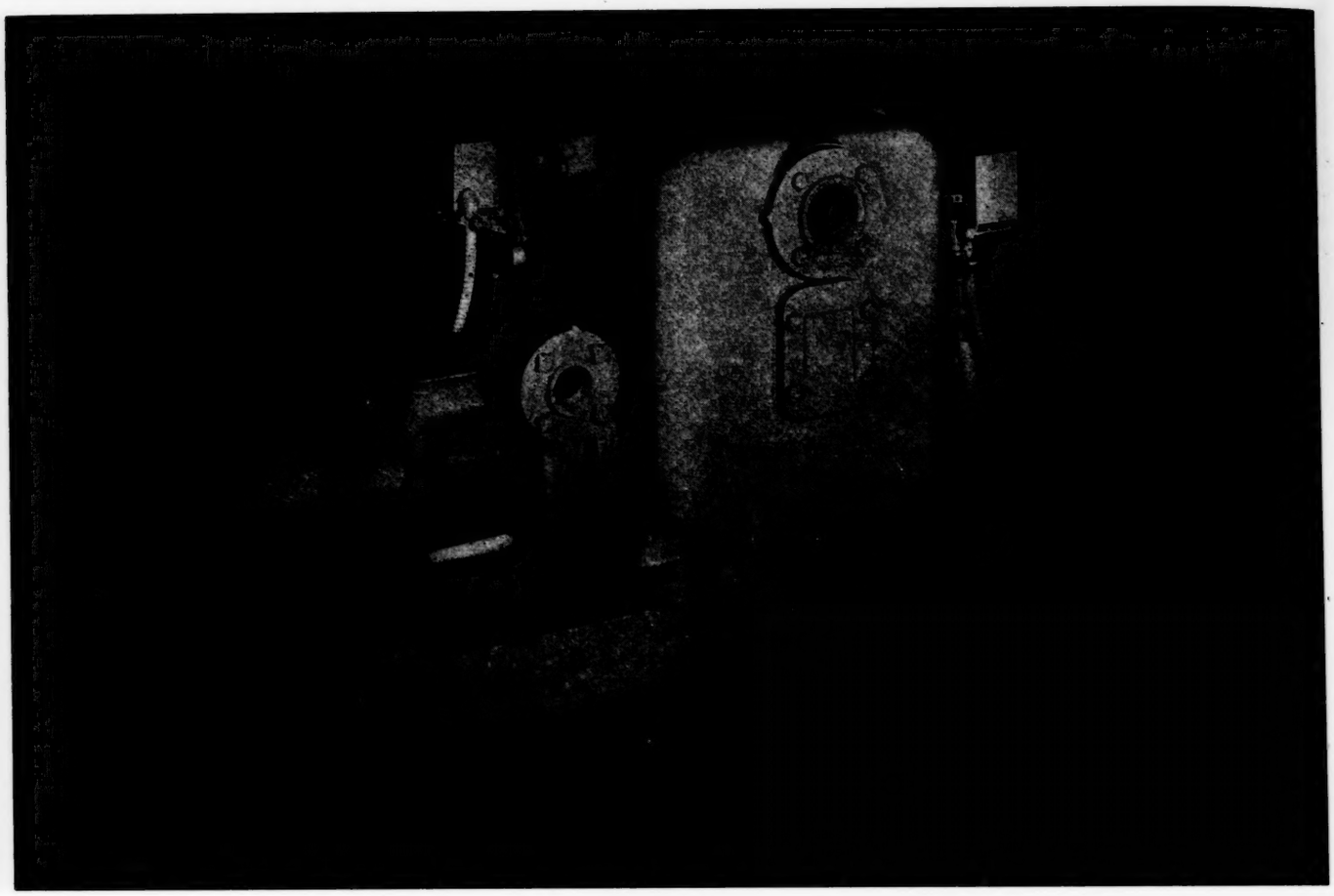

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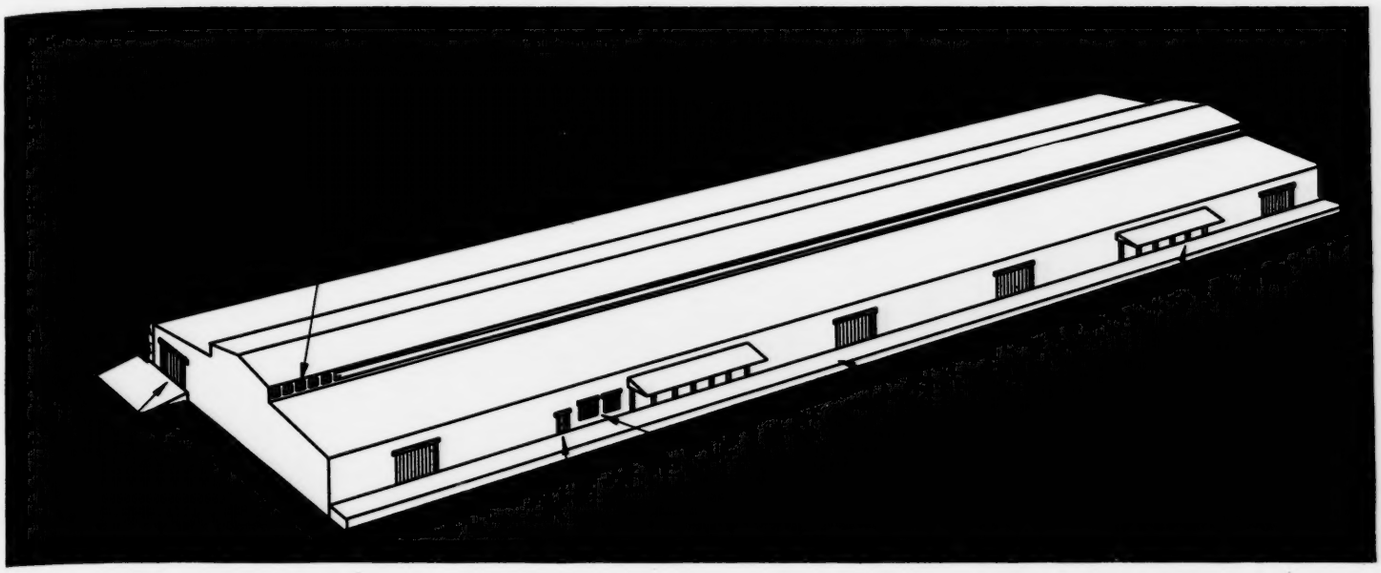


Practical Dehumidification of Warehouses

MAX F. MUELLER
The Davison Chemical Corporation
Baltimore, Maryland

This article will discuss the problems involved in determining the sources of the moisture loads, the methods used to eliminate or reduce them, the practical application of corrective measures to existing structures, correct designs for new construction and the selection of proper dehumidification equipment.

THE dehumidification of large working or storage spaces, such as warehouses, to prevent corrosion and general moisture damage to the stored materials, presents unique problems which are not present in designing a lay-out for comfort air conditioning. In the latter case, summer conditions may require the maintenance of an atmosphere at, say, 60% RH and 75F for comfort considerations and in the winter time heating will be required, possibly with humidification of the air to provide maximum satisfaction. In dehumidification for preservation or industrial reasons, approximately 40% RH is maintained throughout the year, with no regard to the dry bulb temperatures that naturally occur.

It is recognized that rust and corrosion, as well as fungus growth, are reduced to a practically negligible point in atmospheres which are below 40% RH. Unfortunately very little experimental evidence is available which would indicate how high the relative humidity could be before corrosion will start. It is known that each metal or alloy has a critical humidity, which is well under saturation, below which the rate of attack is relatively slow, and above which corrosion proceeds rapidly. Samples exposed for prolonged periods of time at 50% RH in *clean* filtered air showed no visible change and it is now believed that the metal develops an invisible film, which may be an oxide, which has a protective effect. It is definitely known, however, if the average humidity of a space is held below the 40% level that a satisfactory preservation of materials will be obtained. Certain fundamental concepts must be understood before dehumidification can be correctly applied, and are discussed briefly in the following paragraphs.

The water to be removed by dehumidification from a storage space, so that proper preservation conditions may be established and maintained, has two major sources. The first is the excess water initially in the stored materials, which is termed residual water; the second source is the water received from the surrounding atmosphere and which enters the space through breathing or leakage, unless a hermetically sealed space is assumed.

All materials have the capacity to hold and give up water to some degree. Everyone is familiar with the ability of wood to change in water content since changes result in shrinking or swelling of the lumber, a phenomenon which is observed frequently. From this analogy to wood it is easy to understand that all fibrous materials such as cloth, leather and paper would undergo similar moisture changes. In addition, the surfaces of non-hygroscopic materials, such as metal and glass, adsorb water, following the same general laws as those for the more commonly accepted hygroscopic substances. In this latter case, however, the amount of water involved is small. Materials and surfaces may be dried, that is, give up their adsorbed water by subjecting them to an atmosphere having a

low relative humidity, because desorption of water takes place whenever the vapor pressure of the water in the materials rises above the partial pressure of the water vapor of the surrounding atmosphere. Actually, the amount of water in a material exposed to normal weather air is continually changing because the moisture content of the air, and hence the partial pressure of the water vapor, varies constantly and in addition the temperature of the air is also changing.

The amount of water that a given space can hold, as a maximum, varies greatly with temperature, doubling for every 18F to 20F increase in temperature in the normal range. To maintain a fixed percent of that maximum (i.e., a fixed percent relative humidity) over a range in temperature would therefore also require doubling the actual water content of the space for each 18F to 20F increase in temperature. To put this another way, increasing the temperature of air 18F to 20F without increasing the moisture content would cause the percent relative humidity to drop to half the original value. In corrosion control, this result of an increase in temperature would be of no concern, since if a satisfactory percent relative humidity existed before the temperature change, the lowered percent relative humidity following the increase would be even more safe. However, if an equal temperature drop took place, from a safe initial percent relative humidity to double the original value, at that level there would be at least a danger, if not a certainty, of corrosion. This is the usual conception of the complete phenomenon accompanying temperature change in preservation by dehumidification, and in the past it has been common practice to supply sufficient excess dehumidification capacity so that the materials will be safe at the lowest anticipated temperature.

Vapor Pressure

Fortunately, the foregoing description, while true, is incomplete. The change in vapor pressure of the water in stored materials accompanying temperature changes compensates for the change in relative humidity to approximate an isopsychric condition in the atmosphere. An isopsychric condition is defined as one in which there is no change in relative humidity, regardless of the amount of temperature change. As the temperature drops, the cooling of stored materials causes them to adsorb additional water from the surrounding atmosphere. This reduces the partial pressure of the water vapor in the atmosphere and holds the relative humidity close to an isopsychric condition. In practice, a drop in temperature in a space will usually result in a temporary increase in percent relative humidity, since the temperature drop must first take place in the atmosphere before the materials can cool and the water vapor flow between materials and atmosphere cannot take place until after the materials have changed temperature. In a warehouse, where a tremendous volume of space is involved, this effect is not as noticeable as it is in a small package or container; because the large amount of stored materials act as a temperature leveler, to produce a fly-wheel effect.

To illustrate the application of dehumidification to a large space, the discussion which follows will show the problems connected with a large warehouse, say, of 120,000 sq ft of floor space and a gross volume of 3,000,000 cu ft. Assume this warehouse to have a concrete floor, walls of tile and brick construction and wooden roof sheathing supported by wood trusses on wood posts. The interior may be divided into six bays, separated by tile firewalls, in each of which are two large doors. A typical warehouse is illustrated at the head of this article and in Fig. 1. In the example which follows, a 30% RH level has been assumed (rather than 40%) to provide a suitable safety factor.

The main sources of the dehumidification load are the residual water which exists in the stored materials, and the water constantly being received from outside the warehouse, which may come from breathing, infiltration or by transmission.

The residual water exists in the building structure, wooden packing cases, and dunnage used in packing materials inside the packing cases. The amount of lumber in the building structure of our hypothetical warehouse would be in the neighborhood of 29,000 cu ft, or 870,000 pounds of wood.

The weight of wood in the packing cases stored in the warehouse will be dependent upon the method of storage, the height of storage, the aisle arrangement, and the average size of containers stored. For present purposes, it is assumed that 75% of the floor space is utilized for storage. On storage height, the maximum possible height of storage would be within 3 ft of the roof so that sprinkler heads and light fixtures are cleared. Combining all the restrictions, the total usable storage volume would be 1,716,000 cu ft of the total internal volume of 3,000,000 cu ft. However, the net storage space for actual materials in packing cases is somewhat less. Stores are usually placed on 6-inch high pallets and the total volume of space occupied by these pallets would be in the neighborhood of 162,000 cu ft.

Another justifiable loss of space is encountered in the looseness of packing. This will average about 8% of the gross storage space which amounts to 137,000 cu ft. Therefore, the net storage space for actual materials in packing cases would be 1,716,000 — (162,000 + 137,000), or 1,417,000 cu ft net storage. Moisture content of the wood and dunnage in the building may then be calculated as follows:

Weight of wood in building members	870,000 lb
Weight of wood in packing cases	7,085,000 lb
Weight of wood in pallets	1,220,000 lb
Weight of dunnage in packing cases	1,417,000 lb
Total weight of Wood and Dunnage	10,592,000 lb

(The weight in wood in storage spaces averages 5 lb per cu ft of dry wood and the dunnage 1 lb per cu ft of space. Wood in the building structure and in pallets averages 30 lb per cu ft of wood.)

The moisture content of these materials will be 12.5% of the dry weight of the materials, assuming the previous storage to have been in equilibrium with a 70%RH, 70F, temperature condition. When dehumidified to 30% RH and 70F, the moisture content will have been lowered to 6.5% and the weight of

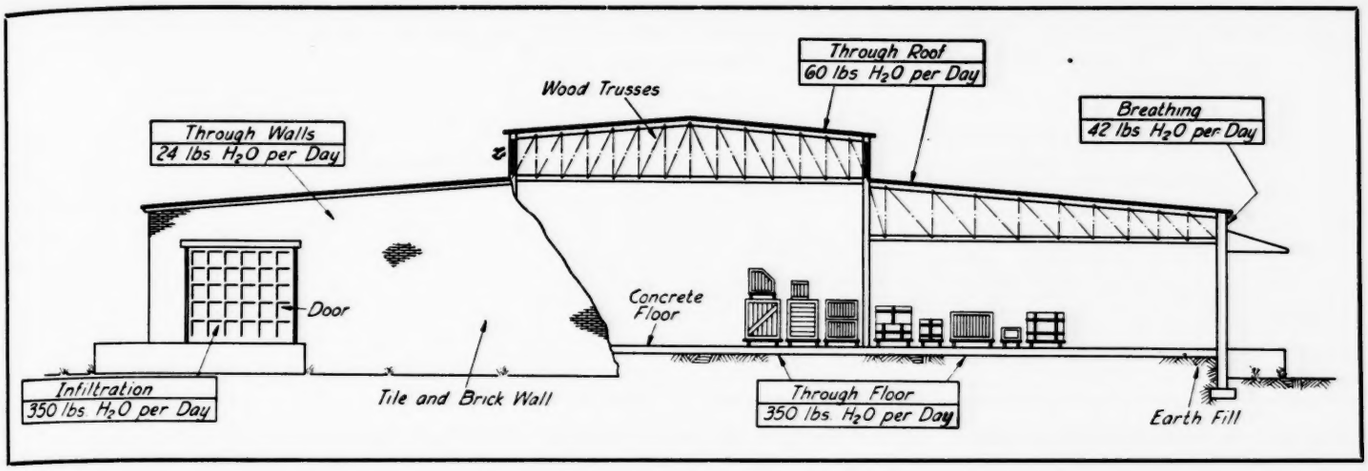


Fig. 1. Cross section of typical warehouse shown at the head of this article, with moisture loads for dehumidification calculation.

residual water that must be removed to establish equilibrium with a 30% RH in the wood and dunnage in the building is therefore 6% of 10,592,000, or 635,520 lb.

Transmission Loads

An average rate of water vapor transmission through the concrete floor has been found by experiment to be approximately 2.9 lb of water per 1,000 sq ft of floor per day, when a 30% RH atmosphere has been maintained above the floor. On this basis, the rate of moisture transmission through the entire floor would be in the neighborhood of 350 lb per day. Experiments have shown that the rate of water transmission through the untreated wall of our warehouse, brought about by maintaining a 30% RH atmosphere on the inner side of the wall, is somewhat less than 4 lb per 1,000 sq ft per day. This value would be reduced to about 0.3 lb per 1,000 sq ft per day by the application of an asphalt vapor resistant sealer. Since this is a laboratory figure, and the commercial application is always less than perfect, it is advisable to use a figure of 0.5 lb per 1,000 sq ft per day for estimation purposes. On this basis, the daily moisture input through some 48,000 sq ft of outer wall would be 24 lb per day.

From standard moisture vapor transmission data of tar and asphalt materials, it is calculated that the roof materials, if in good shape, would have an average moisture vapor transmission rate of less than 0.2 lb per 1,000 sq ft per day. As a safety factor to counteract imperfections in the commerical roof, a higher value of 0.5 lb per 1,000 sq ft per day is assumed. On this basis, the transmission rate through the 120,000 sq ft of roof would be 60 lb per day.

Breathing and Infiltration

Breathing need be considered from the standpoint of temperature change only, as the variations of barometric pressure are too minor in magnitude to make them a factor. The breathing from an afternoon swing of 0.1 inch Hg would result in an air change of only 0.3% of the volume of the building.

Temperature variations inside a warehouse of this character are greatly lessened by the heat absorbing capacity of the materials. Although the temperature in the space just under the roof will change through 60 to 70 degrees on the worst days, the average atmospheric temperature throughout a closed warehouse will not change more than 15 degrees in any one day. Inasmuch as a rise of 5 degrees in temperature forces approximately 1% of the air out of the building, a daily swing of 15 degrees in the average temperature in the building would induce a breathing of 3% of the internal volume of the building, or 90,000 cu ft of air per day. (These calculations, and those which follow are based on the gross volume of 3,000,000 cu ft, for the sake of simplicity.) Assuming the average condition of 70F, 70% RH air entering the building, and 70F, 30% RH air leaving the building in this breathing operation, the amount of water added to the atmosphere in the building through breathing would be 42 lb per day. It is to be noted that this would be the rate for a normal summer day, and that the rate in winter would be much less.

Infiltration from wind pressure can only be an estimate. It is believed that sealing measures would eliminate all major points of infiltration, and thus might reduce infiltration to a negligible figure. For estimating purposes, however, based on available information, a choice of 25% of the volume of the building as daily infiltration is made. Assuming the same ambient conditions as used under breathing, the amount of moisture entry daily from infiltration would be about 350 lb per day.

Summary of Dehumidification Load

The average daily load transmitted into the building is summarized in the following table and shown graphically on Fig. 1.

Moisture Source	Transmission, lb per day
Floor	350
Walls	24
Roof	60
Breathing	42
Infiltration	350
Total daily load	826

This daily load assumes that the interior of the warehouse is being maintained at 30% RH with ambient weather at 70F, 70% RH. Actually, a 40% RH level is satisfactory for most purposes, and this condition will be attained gradually during the first months of operation, the above rates will not be immediately effective when dehumidification is first started.

The given values make no allowance for personnel working in the space, doors being opened, and other sources of artificial water load, to which the following data apply, and must be considered for each case.

A workman, at normal heavy manual labor, releases approximately one pound of water per hour to the surrounding atmosphere. It is emphasized here that low humidity air is perfectly healthy, and no ill effects result from working in these conditions.

A gasoline driven lift truck produces 12 lb of water per gallon of gasoline consumed. In this connection, the monoxide produced by these machines may prove a hazard in a closed building. Therefore, for steady operation only electrical lift trucks should be used.

A 10 ft x 10 ft door opened wide, with a 5-mile wind blowing directly at the opening could admit over 1,000 lb of water to the building in an hour's time on a summer day. It would not normally do so, as the internal pressure would build up and slow the entry after the first few minutes; however, outer doors should never be opened unless absolutely necessary, and should remain open as short a time as possible. Applying conclusions reached by Simpson in Refrigeration Engineering (1936), air admitted per hour of door opening could be as high as 26% of the building volume.

The values given above for the residual water load are characteristic of a fully loaded warehouse. In case the building is not completely filled, the amount of residual water would naturally be less than the above calculated amount. Also, in case there is a considerable removal or addition of materials, the amount of residual water and the time necessary to establish and maintain proper dehumidification conditions will be affected. If materials are moved into the warehouse from a heated warehouse in the middle of the winter, the amount of residual water added with them will be less than if the same materials were transferred from outdoor storage. A warehouse that has a considerable proportion of its materials in bin stock will have less residual water load than one with boxed stores.

Building Modification and Equipment

In preparing and equipping a warehouse or other structure for long term preservation, there are four major field operations to be performed; sealing the building against the ingress of moisture from either vapor pressure differential or windage; providing adequate electric service to operate the dehumidification machines; installing the dehumidification equipment, consisting of dehumidification machines and controls, necessary duct work, and internal wiring; and fourth, arranging for efficient operation.

If it were possible to hermetically seal a building so that air could not enter or leave it, and moisture could not pass through the walls, roof, or floor, it would be a very simple matter to maintain all materials inside in a perfect state of preservation. In such an idealized case, it would be necessary only to dry the materials stored in the warehouse to the desired level, then close the warehouse until such time as the materials were needed. Since the amount of water in the warehouse would not have changed, no moisture damage would have occurred in the interim. Unfortunately, even if it were possible to so seal a building it would still be impractical to do so. Temperature and barometric changes would cause relatively large pressures and vacuums to be set up as the weather varied, and the walls of a hermetically sealed building would eventually rupture unless it were built with internal rigidity akin to that of a battleship. In structures such as normal warehouses, even the best workmanship and materials will not seal the building so tightly as to stop normal breathing. It is advisable to seal the apertures between the interior of the building and the outside as tightly as economically feasible.

The most common sources of infiltration are the following: cracks around outside doors; cracks around windows in monitors and side walls; and openings in the roof for ventilators and manholes. The treatment of these sources of infiltration is relatively simple and inexpensive, consisting of closures of lumber, or caulking materials, as the case warrants.

Mechanically closing such openings against excessive infiltration does not properly complete the treatment of the external structure. Moisture will find its way continuously through the walls, the floor, and the roof, even in clear, dry weather, due to the difference in vapor pressure between the inside and outside of the building. The amount of moisture that will so pass through these points depends upon the difference in vapor pressure between the inside and the outside of the building, and the vapor transmission rate of the materials.

This rate of transmission can be cut down by coating the interior walls of the structure with a vapor barrier. In the case of the floor, a surface coating is not recommended, since the surface coatings available would be incapable of resisting the inevitable abrasion. The cost of treating the underside of the roof would not be economically justified. The normal roof materials (tar, bonded in place by paper) are in themselves highly resistant to the passage of moisture vapor, and little, if any, gain would be obtained from adding an additional coating of a similar material.

The material used for coating interior walls of the building is an asphaltic or bituminous compound which is applied in a thick coat with a spray gun in the same manner as spray painting. The material clings to metal, wood, ceramics, and glass, provided the surface is not initially water-soaked or coated with oil. A single application is sufficient for the purpose, leaving a coating that is approximately 1/16 inch thick when applied, and shrinks to 50% to 80% of this thickness as the solvent carrier evaporates. The final dry coating is firm, but not hard. By this treatment the average moisture vapor transmission rate is reduced to less than 0.3 lb per 1000 sq ft per day.

Since the inner side of the monitor windows will be covered by the dark colored moisture sealing coating, it will be necessary to paint all windows on the outside

with a light tinted paint, to reflect heat. Otherwise, the trapping of heat brought about by the dark interior coating would cause the glass to crack.

Certain external doors would have to be left unsealed for access to transfer materials, but the number should be kept at a minimum, not over three per warehouse and preferably only one. For personnel access, one normal size swinging door would be left unsealed at each end of the warehouse.

Inside the warehouse, no sealing would be required on any of the sliding doors in the firewalls. Both such doors would be arranged as shown in Fig. 2. Closing doors in this manner would serve as a sufficient control of the path of the air movement in the building.

Dehumidification Machines and Auxiliary Equipment

Dehumidification of a warehouse to the desired level of humidity will require the initial removal of the large amount of residual water discussed above plus the daily load which will vary with the seasons. As the residual water is gradually removed, the daily input will become greater, due to the ever-increasing difference in the amount of water in the air of the warehouse and that of the surrounding atmosphere.

It would be impractical to attempt to attain full removal of the residual water in the first month, or even in six months, inasmuch as the materials release their water at a relatively slow rate and no particular advantage is gained by attempting to hurry the process. A study of all the factors involved indicates the optimum time interval in which to strive for complete establishment of the 40% RH conditions, with a complete removal of residual water, to be approximately one year.

The total capacity of the dehumidification equipment required to attain the desired result in a large building in one year should be divided among a number of dynamic machines which would be caused to operate by humidity reactive control devices whenever the conditions exceeded the desired level of humidity. These machines should be self-contained units of the dual-bed solid adsorbent type, complete with the necessary fans, motors, heaters, valves and electrical equipment. In this type one bed is being reactivated while the other is adsorbing, and the operation is entirely automatic. For this service, a characteristic dehumidification machine performance table is as follows:

Entering Air Conditions			Water Removal, Pounds per Day
Dry Bulb, F	Dew Point, F	% RH	
70	41	35	500
55	28	35	300
30	6	35	115

The entering air conditions shown are the usual design values which are common in the dehumidification equipment industry. The latest design practice also limits the air velocity through the gross area of the center of the desiccant bed to 60 feet per minute and specifies that the direction of air flow during reactivation be counter to that during adsorption. The desiccant beds in this type of machine are normally four to six inches thick, and the air must pass through a filter to remove dirt and foreign matter before being blown through the desiccant.

In order not to overload the power requirements it is advisable to limit the electric energy demand of the above machine to 25 kw of 3-phase current, divided between the fan motors and the reactivation heaters. The control equipment may be operated on 110-volt, single-phase, 60-cycle current.

After the first year of operation, during which the residual water has been completely removed, the dehumidification equipment need only counteract the daily receipts of moisture. This will result in operation of properly selected equipment less than 50% of the time.

Dehumidification machines operate properly when distributed throughout a warehouse without the necessity of an extensive duct system by utilizing the existing internal structures. They must be located along outside walls, to permit installing intake and discharge

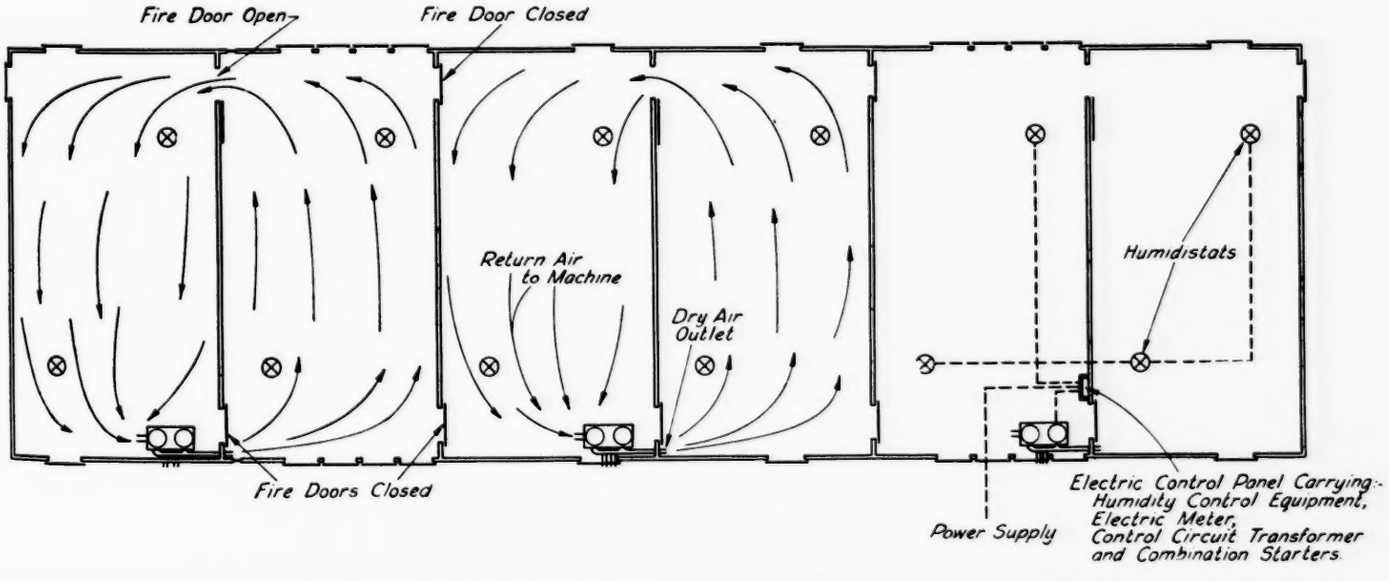

Fire Door Open
Fire Door Closed
Return Air to Machine
Dry Air Outlet
Humidistats
Fire Doors Closed
Power Supply
Electric Control Panel Carrying Humidity Control Equipment, Electric Meter, Control Circuit Transformer and Combination Starters

Fig. 2. Plan of warehouse showing location of dehumidifying machines.

connections to the outside atmosphere for reactivation. A typical layout is shown in Fig. 2. Fig. 3 shows diagrammatically the dehumidification machine.

Control is obtained through a multiple hook-up of a number of humidistats which cause the machines to operate whenever the humidity is above the desired operating level and stop them as soon as satisfactory conditions are restored. A satisfactory control for a 3,000,000 cu ft building would require twelve humidistats, connected to relatively simple associated control panels. Temperature and humidity recorders are necessary in order that the results obtained from operation may be checked. Not more than two hygrothermographs per building would be required for this purpose.

Since hair element humidity devices are subject to characteristic inaccuracies, facilities are required for periodically checking their accuracy. A test room for this purpose is recommended at each station, but one such room would serve any number of dehumidified warehouses. Equipment for the test room consists of means to cycle the room through a range of humidity, an accurate humidity and temperature recorder against which to compare operation, and operation recorders to permit the checking of humidistats to continue unattended. Tests of each humidistat and hygrothermograph should be made at intervals of not greater than two months. One permanent operator can handle two dehumidified buildings.

New Construction

The preceding discussions have been concerned with what must be done to an existing building to prepare it for dehumidification. There is also the possibility that a new building may be constructed solely for use as a dehumidified warehouse, in which case certain engineering modifications are justified.

Contrary to normal expectations, expensive additional construction is not necessary when building a warehouse to enable it to be satisfactorily dehumidified. Preliminary tests and calculations have shown that the difference in the annual cost of operations for dehumidification between a specially constructed vapor proofed building and a normal warehouse, which has been properly vapor proofed as outlined, does not justify much additional expense in construction.

The breathing load cannot be cut down for the reason that it would be impossible and undesirable to seal a building hermetically. Further, even with the best of construction it would not be practical to reduce the normal water receipts through the structure by more than 1/3 of those into a properly sealed normal building. It is impossible, therefore, to save more than 1/3 of the power cost of operating dehumidification machines. Since the annual operating cost for power to dehumidify an average 3,000,000 cubic foot building will be on the order of $3,000, the saving thus attained is only $1,000 per year. This amount, amortized over, say, twenty years, would not justify much additional construction.

The following items are considered justified for incorporation in the design of any new construction:

(a) A concrete mix should be used which would provide a floor with a lower water vapor transmission rate than that of a normal concrete floor.

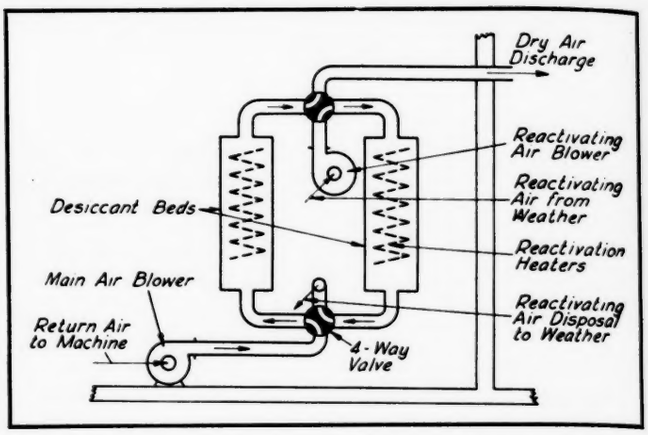


Fig. 3. Diagrammatic sketch of dehumidifying machine.

(b) Brick or concrete construction should be used for the side walls rather than wood. These should then be vapor proofed by applying the sprayed materials discussed previously to the inside at the time of construction. This will produce a satisfactory vapor barrier at far less cost than attempting to incorporate the water vapor barrier in the wall construction.

(c) The building should be built without windows, with the exception of those required in offices.

(d) The buildings should be designed so that all apertures to the outside, particularly at the eaves, are sealed with a material which will not shrink and produce cracks. One method of doing this would be to design the normal roofing to permit an asphaltic type seal between the roof and the side walls. Another would be to use copper flashing to fit between the wall and the roof.

(e) The roof, if built of standard 5-ply roofing materials, will be satisfactory, and requires no additional treatment.

(f) All doors should be weatherproofed against infiltration of ambient air. Door opening meters will reduce the moisture admission from that source.

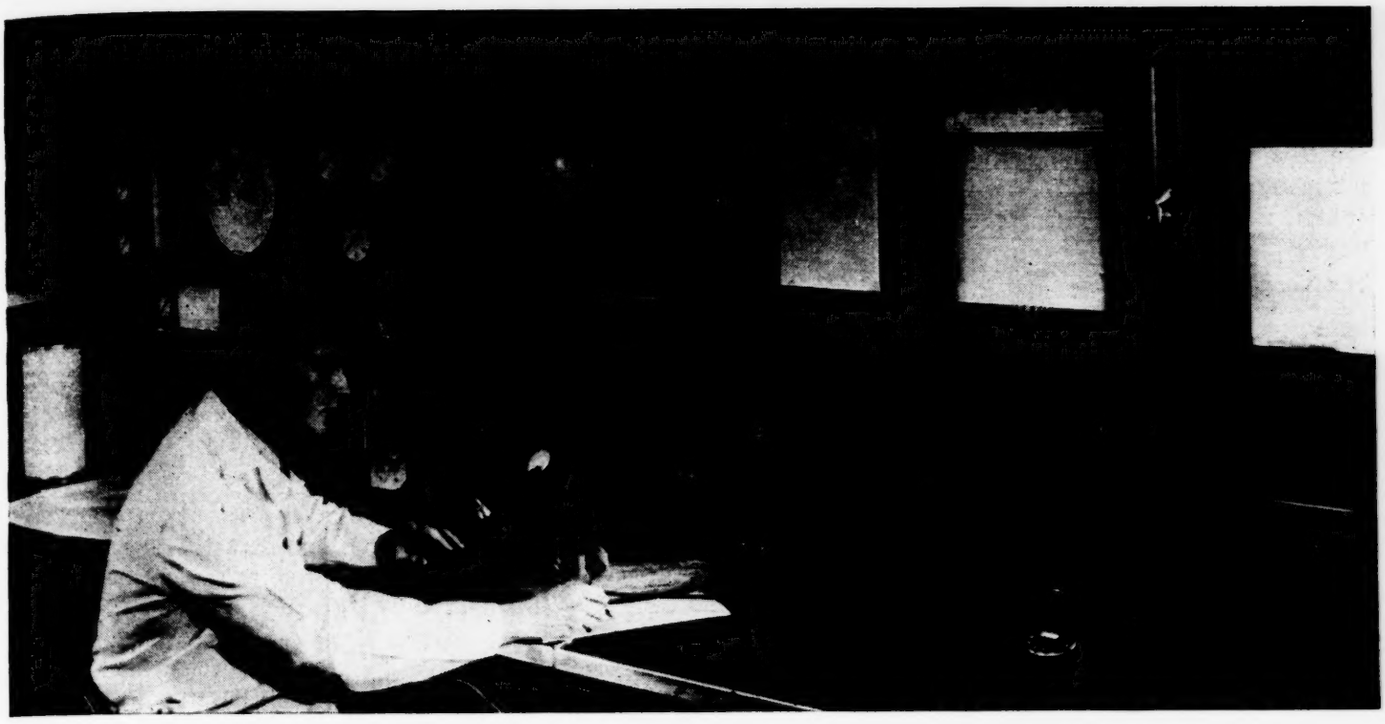
(g) Any personnel spaces, such as offices and lavatories should be vapor sealed from the warehouse space.

(h) Dry sprinkler systems for fire protection should be used.

Incorporating these ideas into the design of the new construction will provide all the practical requirements which make dehumidification of the warehouse feasible. The expense of these additions to the design will not be more than that required to prepare an already existing structure for dehumidification.

Conclusion

The preceding discussion has illustrated the inherent problems that are met in preparing a building for dehumidification and in dehumidifying it. The reader is reminded that the figures used will not apply to all cases. The residual water loads will depend upon the interior materials of construction of the warehouse or other building under study, the type of materials to be stored therein and the level of humidity desired, and must be calculated for the particular situation.



Control panel board of the partitional calorimeter at the John B. Pierce Laboratory of Hygiene.

Effect of Thermostat Characteristics on Panel Heating Performance

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John B. Pierce Foundation, New Haven, Conn.

Experiments were undertaken at the John B. Pierce Foundation Laboratory of Hygiene to determine the effect of thermostat characteristics—that is, enclosure type and whether on-off or modulating – upon the operation and performance of a low heat capacity electric panel heating system. In a two-part article, of which this is Part 1, the authors discuss significant differences between convective and radiant heating with respect to control principles and show the effect of radiant and convective components upon comfort and thermostat selection criteria.

PHYSIOLOGISTS have shown (1 and 2)[1] that when the mean radiant temperature of an enclosure is *reduced*, the occupant loses an increased proportion of of body heat by radiation. In order to compensate for this increased radiation loss it is necessary to reduce the convective loss by increasing the air temperature. In general, comfort equivalent to that obtained when enclosure and air are at the same temperature (70F may be taken as the basal condition) may be obtained by increasing the air temperature one degree Fahrenheit for each degree reduction in mean radiant temperature. This relationship holds reasonably well for any enclosure in which the air velocity is low—less than 35 fpm. It has also been shown that when the mean radiant temperature of an enclosure is *increased*, the occupant loses less heat by radiation, and it is therefore necessary to increase the convective loss for equivalent comfort. This may be accomplished by reducing the air temperature.

The first case above applies to the convective types of heating system, i.e., radiator, convector, and warm air types. With these systems, as the outside temperature becomes colder, the inside surfaces of all the exposed areas also become colder, the mean radiant temperature is lowered, and a higher air temperature is the requirement for equivalent comfort. With radiant heating systems, however, the second case applies. A reduction in outdoor temperature results in lowered inside surface temperatures of the exposed areas, but the increased demand for heat results in warmer heating panels. The net result is an increase in the overall mean radiant temperature with a consequent requirement for a reduction in air temperature if equivalent comfort is to be maintained.

The above discussion will serve to show the primary difference between convective and radiant heating systems with respect to the control problem. In short, a convective system requires an increase in air temperature while a radiant system requires a reduction in air temperature with increasing load.

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[1]Numbers in parentheses indicate references listed at the end of Part 1 of this article.

Since comfort with either convective or panel systems is a function of both air temperature and mean radiant temperature, it appears reasonable that a suitable thermostatic control of either system should be sensitive to both factors, and should be capable of producing equivalent comfort with varying load without the necessity of resetting the instrument.

The Globe Thermometer

The globe thermometer is probably the simplest and best instrument for integrating air temperature, radiant temperature, and air movement. It consists of a hollow copper sphere (generally 6 inches in diameter) coated with matt black paint, and containing an ordinary mercury thermometer (or thermocouple) with its bulb at the center of the sphere. When used in conjunction with a shielded air thermometer and a katathermometer, it gives a very reliable indication of the combined effects of the thermal environment on human comfort. The temperature of the instrument depends solely on the environment in which it is placed. If the surfaces which surround the globe are warmer than the air, the temperature recorded by the thermometer inside the globe will be above air temperature; and conversely, with walls and other surroundings cooler than the air, the globe thermometer temperature will be below air temperature. For any given situation, when equilibrium is reached, the temperature of the globe thermometer will be such that the globe's gain or loss of heat by radiation must be balanced by the loss or gain by convection to the surrounding air. In still air (15 to 25 fpm), which is ordinarily encountered in residential heating work, the globe thermometer placed approximately in the center of a room is the most acceptable and representative single measurement of room thermal conditions, and most closely approaches measurement of human thermal sensation.

By reference to the nomograph of Bedford and Warner's paper (3) on the globe thermometer, it can be shown that, for an air velocity of 25 fpm, when the air temperature and the mean radiant temperature are varied in equal and opposite directions above and below 70F, the temperature indicated by the globe thermometer remains essentially constant at 70F. This fact is indicated in the tabulation below for both the convective and radiant situations:

Air Temp., F	Globe Temp., F	Mean Radiant Temp., F
	For Convective Systems:	
70.0	70.0	70.0
71.0	69.9	69.0
72.0	69.8	68.0
73.0	69.8	67.0
74.0	69.7	66.0
75.0	69.7	65.0
	and For Radiant Systems:	
70.0	70.0	70.0
69.0	70.1	71.0
68.0	70.2	72.0
67.0	70.3	73.0
66.0	70.4	74.0
65.0	70.6	75.0

On the basis of this important characteristic, the globe thermometer may be taken as a measure of equivalent comfort (for the heating condition with low air movement), and consequently this instrument was used as the primary criterion of performance of several types of thermostats and thermostat covers.

In addition to constancy of the average globe temperature, other factors of importance in judging control performance are variations in temperature of the heating medium (panel heated ceilings in this case), cycle duration when on-off controls are used, the maximum-minimum differences in globe temperature, and possible fuel savings.

Object

The primary object of the experiments reported was to determine the requirements of a thermostatic control which would be most suited to the optimum operation of low heat capacity panels utilizing electricity as the energy source. The tests performed were all restricted to the panel heating situation, and the introductory comments on the comparison between panel and convective systems have been presented to provide a more generalized background for interpretation of the test data.

Description of Test Structure

The experiments described hereinafter were performed at the John B. Pierce Laboratory of Hygiene in the experimental structure which consists of a two-room house surrounded by shell spaces on all four sides and above the roof and below the floor. In these shell spaces, temperatures may be maintained at any desired point to simulate external weather conditions. The structure is one-story in height, and the two rooms are separated by a 4-ft wide interior corridor which is used as a control room.

Only one room of the experimental house was used for the experiments described. This was the so-called East Test Room which has exterior walls facing north, east, and south, and the one interior wall adjacent to the control room facing west (see Fig. 1). The room is 15 ft long by 12 ft wide and 7 ft 10 inches in height, and has two windows in the east wall and one in the south wall. The windows are double-hung with metal weather stripping, are 2 ft 6 inches by 4 ft 8 inches in size, and have storm sash, weather stripped with $\frac{1}{8}$ inch felt. The exterior walls are of the pre-fabricated sandwich type construction and consist of 2 inches of cellular glass bonded between $\frac{1}{8}$-inch plywood sheets. (Overall Coefficient of Heat Transmission = 0.26 Btu per (hr) (sq ft) (F).

The heating requirement for the test room was approximately 2,300 watts for the steady heat flow condition with a 70F indoor-outdoor temperature differential. This heating requirement was satisfied by means of a panel heated ceiling of the type[2] which consists of natural rubber to which has been added specific materials permitting the conduction of electricity. The rubber provides a uniform heating surface over the

[2]Manufactured by the Uskon Section, Wire and Cable Department, United States Rubber Company, New York, N. Y.

entire area. The conductive rubber layer is sealed between layers of phenolic impregnated insulation, and the laminated construction is made rigid by a backing of 3/16 inch asbestos cement board. The completed bonded assembly is approximately ¼ inch in thickness. In order to keep the panel temperatures to a minimum, full ceiling coverage was used, and this consisted of fifteen 3 ft x 4 ft panels. To satisfy the above heating requirement, the average watt density was approximately 13 watts per square foot, and the maximum was 16 watts per square foot. Fig. 1 is a plan of the test room and the shell spaces, and shows the arrangement of the ceiling panels.

Thermal measurements were obtained primarily through the use of iron-constantan thermocouples in conjunction with continuous type chart recorders. Temperatures were recorded at 15-minute intervals from the following locations: interior wall, floor and ceiling surfaces, air 6 inches from the surfaces, air and globe in the center of the room at the 48-inch level, at or within the thermostat being used for a specific test, and all outside shell spaces.

Power measurements were obtained by the use of ammeters and volt-meters for tests in which on-off types of control were utilized. The durations of on-off cycles were obtained by a recording watt-hour meter which was also used to indicate energy consumption in tests which utilized modulating type thermostats.

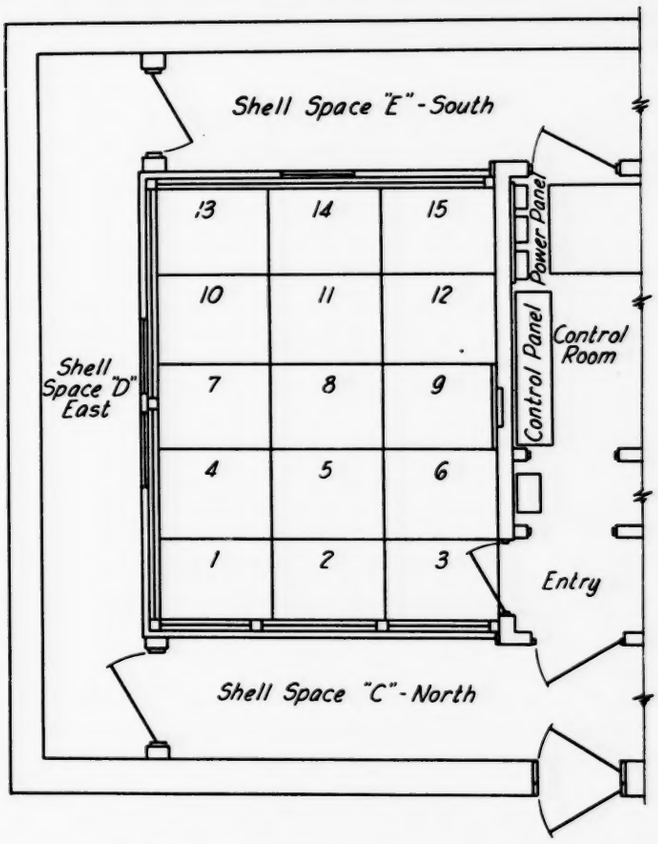

Fig. 1. Plan of test room and shell spaces.

Criteria for Design of Tests

In the design and execution of the tests described below, several factors affecting performance were considered. These were: (1) thermostat operating differential, (2) type of thermostat, i.e., on-off or modulating, (3) cover design and finish, (4) location, and (5) load variation. No attempt was made to rate thermostats made by various manufacturers on a competitive basis. For the tests performed with the on-off type of control, a single thermostat was used, with several cover modifications, thus making it possible to study the effects of cover design and external factors with no variation in thermostat mechanism. This thermostat was of the type employing sealed-in-glass mercury contacts and was selected primarily because of its low thermal capacity, quiet operation, and ease of differential adjustment.

The tests to be described may be classified in several groups which will be considered separately and which are roughly synonymous with the listing of factors affecting performance given in the paragraph immediately above.

Group 1 – On-Off Control

This group consisted of two tests in which the only variation was the thermostat differential. In one test, the total differential was 3.5 degrees and in the other test it was 1.0 degree. The thermostat was of the simple on-off low voltage type and controlled, through an intermediate transformer relay, an input of 1,800 watts to the ceiling heating panels. Location of the thermostat was on the interior partition. Exterior environment was held at 20F, thus providing approximately a 50F interior-exterior difference with the thermostat set for 70F.

Table 1 presents significant data for the two tests of thermostat differential. These data were derived from three complete on-off cycles during the periods of thermal equilibrium. Most significant are the reductions in the maximum-minimum differentials which may be attributed to the reduction in cycling time produced by the smaller thermostat differential. This is extremely important since it greatly reduces the high temperature peaks caused by excessive ceiling temperatures, and the low temperature troughs produced by surfaces and air cooling simultaneously. A 3.5-degree variation in temperature within the thermostat resulted in a 5.0-degree (65.5F to 70.5F) variation in the 48-inch globe in the room center, while a 1.0-degree variation at the thermostat caused a 1.3-degree (68.7F to 70.0F) differential in the room center.

TABLE 1.—EFFECT OF THERMOSTAT DIFFERENTIAL

Item	Differential, F	
	3.5	1.0
Average Globe Temp. at 48-in. level, F	68.0	69.2
Minimum Globe Temp., F	65.5	68.7
Maximum Globe Temp., F	70.5	70.0
Maximum-Minimum Difference for Globe Temp., F	5.0	1.3
Maximum-Minimum Difference in Ceiling Surface Temp., F	24.5	9.0
Length of 1 complete cycle, minutes	149	50

Group 2 – Modulating Control

A single test was performed to show exactly what might be expected of a modulating control when operating under the same conditions as the on-off control. The control selected was of the low voltage type and operated a reversible low voltage motor which was directly connected to the shaft of an induction type voltage regulator. It was thus possible to vary the ceiling input by voltage variation produced by thermostat dictation. Exterior temperatures were 20F and 45F, and the test room thermostat setting was 70F.

Analysis of the data showed that (1) variations in ceiling surface temperatures for the stable phases were less with the modulating control, being 5.5 degrees for the 20F outside condition and 2.5 degrees for the 45F outside condition, as compared with averages of 8.4 and 8.9 degrees, respectively, for on-off control in four tests, (2) the average maximum-minimum difference for all points (excepting ceiling surface and air 6 inches below the ceiling) and for both exterior conditions was 0.75 degree as compared with 0.85 degree for on-off control.

In general, the performance with the modulating control was slightly superior, although the reduction in temperature variations at the heating surface (ceiling) did not produce significant changes in the maximum-minimum differences for other points. Therefore, when the much higher cost of the modulating control arrangement is considered, it would seem extremely difficult to justify its use with the low heat capacity electric heating panels. In addition, the use of an on-off control in each room of a residential installation provides a flexibility which cannot be approached by modulating control with its inherent necessity for voltage modulation when used with electric panels.

Part 2 of this article will show the effect of thermostat enclosure characteristics on performance of an electric panel heating system.

References

(1) Winslow, C.-E. A., Herrington, L. P., and Gagge, A. P., The Determination of Radiation and Convection Exchanges by Partitional Calorimetry, *Am. J. Physiol.*, Vol. 116, pp. 669-684, 1936.

(2) Winslow, C.-E. A., Herrington, L. P., and Gagge, A. P., The Relative Influence of Radiation and Convection upon the Temperature Regulation of the Clothed Body, *Am. J. Physiol.*, Vol. 124, pp. 51-61, 1938.

(3) Bedford, T., and Warner, C. G., The Globe Thermometer in Studies of Heating and Ventilating, *J. Hyg. Camb.*, Vol. 34, p. 458, 1934.

Danish Studies Probe Heating and Comfort Criteria

Insulation applied to the inner surface of exterior walls saves more fuel than the same amount of insulation located nearer the outside surface. This is especially true where intermittent heating is practiced, according to experiments conducted by a Danish committee for the study of domestic heating. Internal insulation of an existing building will save fuel over and above the saving obtained by reduction of the heat transfer coefficient.

Aluminum foil applied to the interior walls increases convection velocities in a room and decreases heat requirements by 14% as compared with an uninsulated room, the Danish committee reported.

These Danish experiments were conducted by a committee for the study of domestic heating set up at the instigation of the Technological Institute at Copenhagen through the auspices of various Danish philanthropies. The committee consisted of professors, engineers, architects and government officials under the chairmanship of Professor August Krogh of the Technological Institute. Work begun in 1938 was conducted with various interruptions by the German occupation through 1945 and resulted in a series of 12 reports, published in Danish with English summaries.

Cold feet and hands are the chief causes of sensations of being uncomfortably cold, the Danish tests showed. Body temperatures of young women remained constant while skin temperatures decreased with lower environmental temperatures. Body temperatures of young men decreased slightly with less falling off of skin temperatures. Nevertheless, both young men and young women feel "comfortable" under practically identical conditions. Body temperatures of older people fall more rapidly with decreasing ambient temperatures in spite of warmer clothing. Warm floor temperatures do not prevent cold feet if the surrounding air temperature is not adequate. For a rise in floor temperature from 68F to 86F (a rise of 18 degrees) skin temperatures of the ankle rose only 2.2F.

Reflex heating, described as an arrangement of continuous tube along walls which had been covered with aluminum wall paper, showed fuel savings from 14% at 41F outside temperature to 42% at a mean outside temperature of 50F as compared with heating identical actual dwellings by radiators.

Exposure to a cold surface on one side, such as occurs when a person is seated near a single glass window in winter, showed that tissue changes (thickening of the subcutaneous layers) occurred on the cold side of the subject. In spite of this, subjects in the Danish experiments reported that both sides were comfortable, indicating, it was said, that the sensation of comfort is not always a criterion of thermal equilibrium.

Danish experiments to determine the relative radiant and convective heat losses from human beings showed a distribution of 36% by radiation and 64% by convection. These results differed from similar tests at the John B. Pierce Laboratory of Hygiene which showed a heat loss distribution of 55% by radiation and 45% by convection. Both series of tests were conducted with cold air and wall temperatures. Emissivity of human skin was determined to be .985.

Intermittent vs. continuous heating in two blocks of flats for old-age pensioners in Copenhagen showed economies for the intermittent heating of from 4.3% to 8.2% with outside temperatures from 39.7F to 25.5F.

Rain Water Leaders Serve as Ducts for

Forced Air and Radiant Heating System

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Two types of heating systems which employ conventional rain water leaders for ducts are described and illustrated in detail. These designs can be used for small and moderate size basementless type homes.

A HEATING system which lends itself so readily to the basementless house, physically and economically, yet provides personal comfort, is illustrated and described. This system heats the air quickly and directly, and controls air temperature and circulation. Meanwhile, the fan is kept running to eliminate the stratification of cold air experienced in on-off systems. Such a system also affords air cleansing and humidity control, together with a fair amount of radiant heat. A further and incidental advantage may be obtained by means of a fan switch for summer operation, thus enabling one to circulate three or four air changes throughout the house.

The method illustrated is intended for use in small and moderate size compact residences of one or two stories, and also where the farthest supply or return outlet will not be more than 60 ft from the furnace bonnet, measured along the duct. The particular house shown, Fig. 1 and 2, one of a group of ten, does not lend itself as well to the method as would many a more simple structure with centralized furnace. It is shown to illustrate how many problems can be solved. Normally, ducts for the second floor could be all run together just outside the stair hall. Here, they could be furred in a 12 x 8-inch enclosure at first floor ceiling were it not for the low headroom at the door from entrance to living room. This situation was met by running the ducts, Fig. 2, in a couple of false beams.

Investigations by the writer as to what others might be doing in the way of forced air radiant floor and wall systems, led in time to conferences with C. M. toeLaer, engineering department, General Electric Co., Bloomfield, N. J. This department has done a rather thorough job of investigating and developing the system described. All necessary arrangements were made for application of a complete system consisting of furnace, ducts and special high velocity wall outlets. A tabular and graphical presentation gives the maximum heat carrying capacity of each 4-inch round aluminum duct and wall outlet.

These data, arranged according to the length of duct and number of its turns, give pressure drop and heat loss—factors that affect the net delivery of heat to the room. A 10 to 20% allowance is made for heat loss

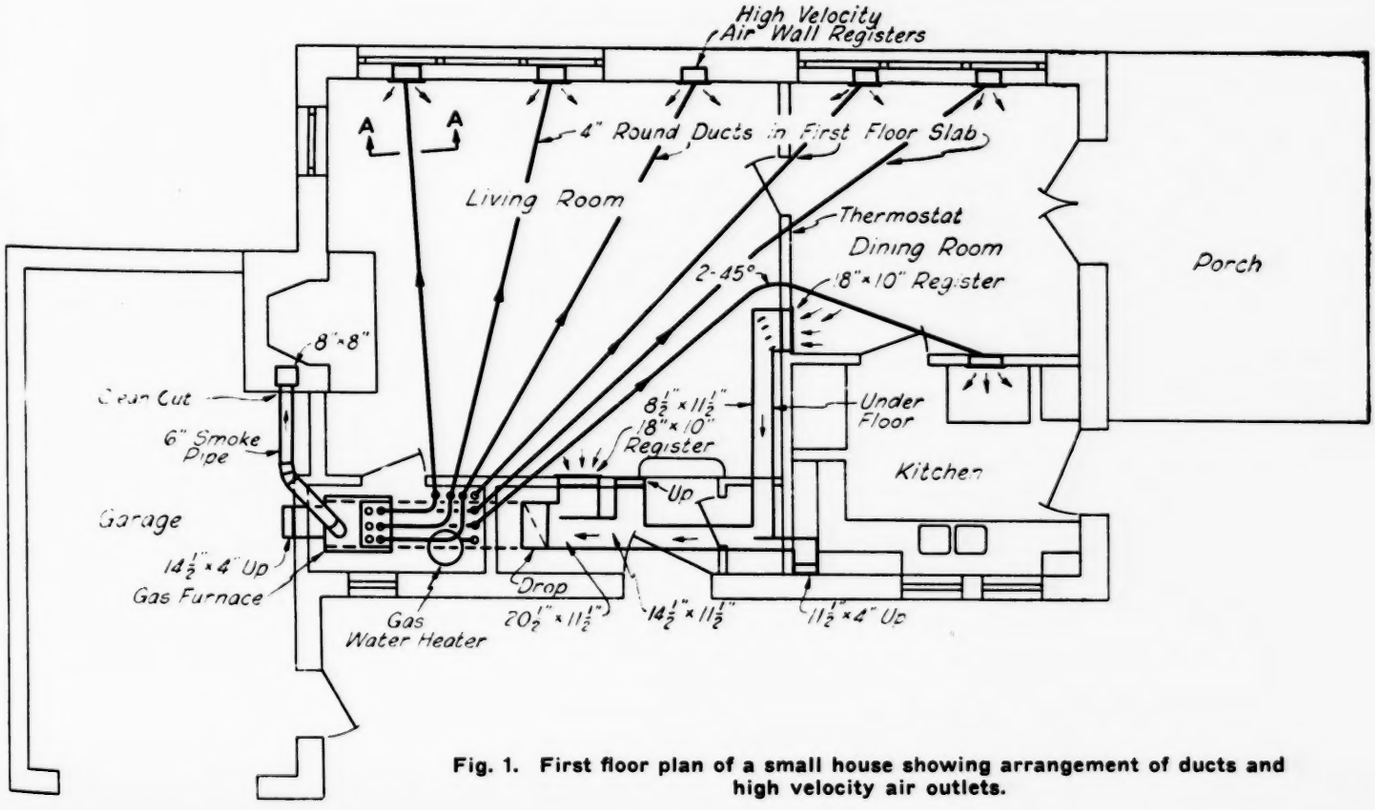


Fig. 1. First floor plan of a small house showing arrangement of ducts and high velocity air outlets.

TABLE 1—COST OF DUCTS TO THE HEATING CONTRACTOR

Size and Material	Cost per Linear Foot
3-inch G.I., 28 gage	$ .11
3-inch G.I., 26 gage	.13
4-inch G.I., 28 gage	.15
4-inch G.I., 26 gage	.18
4-inch drain tile	.13
4-inch B & S tile	.25
4-inch fiber conduit	.23
4-inch aluminum*	.30
4-inch Transite	.45
4-inch x 5½-inch—½-inch thick†	.80
4-inch x 5½-inch— 1-inch thick†	1.02
4-inch rubber hose	2.00

*B & S 26 gage.
†Preformed asbestos duct, 3.3 lb and 6.9 lb per linear foot, respectively.

from ducts according to the conditions under which a duct is to be installed. In general, a 4-inch duct delivers about 64 cfm, which figure sets the velocity at 730 fpm. Therefore, the heat capacity of the duct before heat loss is 64 cfm × 60 minutes × (190F — 70F) × .018 Btu = 8384 Btu. Total pressure drop ranges from 0.35 to 0.40 inches of water and this is about equally divided between the supply and return ducts. Both temperature and pressure drop are less where the furnace can be centrally located, particularly so as this usually requires only a single return register with a short duct connection with few turns.

Locating Outlets for Greatest Comfort

Preferred location for the special warm air grille outlets is on the exterior wall and low down above the baseboard so that the air may be directed upward and radially along the cold glass and wall. This, in a sense, gives the same effect as radiant heating by eliminating or reducing the radiation from occupants to the cold outdoor wall. The warmer air is also delivered in opposition to normal cold air currents within the room and at such velocity and directions that infiltrating cold air is quickly diffused with room air before contacting room occupants.

Supply outlets, could, of course, be located low on inside partitions. Return outlets would then be located in the baseboard of exterior walls under windows. Such an arrangement counteracts a portion of the normal drafts within the room, although it does not offer the comfort advantages of supply outlets on outside walls. The arrangement has a further disadvantage in that more points of air return are required to produce warm air circulation to all areas along the cold outside wall.

Regardless of how few supply outlets are required to offset the room heat loss, there should be a sufficient number of these outlets to give good heat distribution and also to cover the entire exterior wall. Such outlets should be located on the longest outside wall, preferably not closer than 3 ft from adjoining interior partitions and offset from windows, Fig. 5, where drapes interfere. In the case of small rooms, one outlet in the middle of the outside wall will be sufficient.

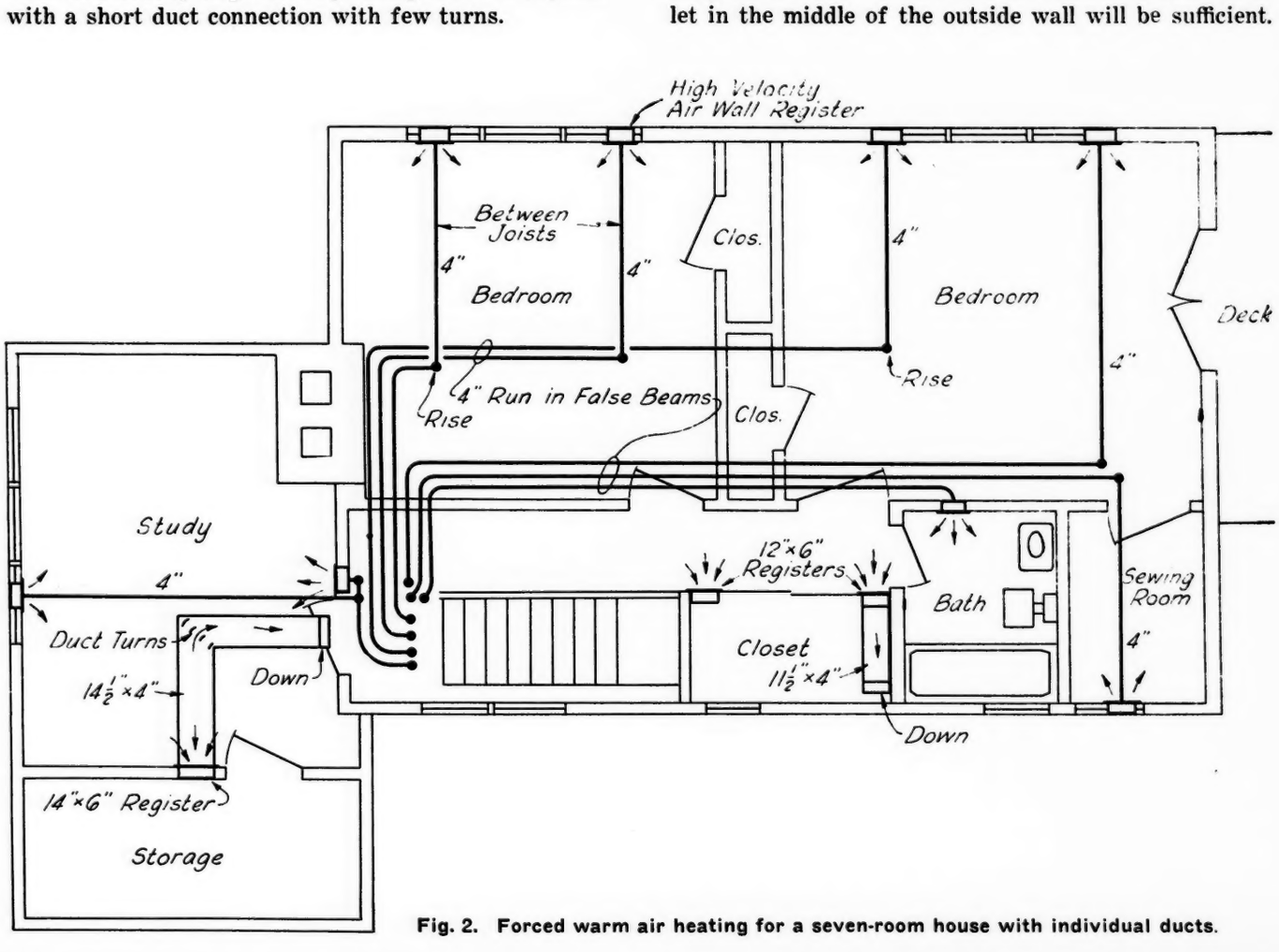


Fig. 2. Forced warm air heating for a seven-room house with individual ducts.

Location of supply outlets above the baseboard simplifies installation. Returns, however, should be in the baseboard with the baseboard carried up and around registers of large size. There are fewer returns than supply outlets. While there should be no return in bath rooms or kitchens, returns from dinettes or breakfast nooks should not be objectionable.

Wall outlets collect less dust than floor outlets, and since the discharge is radial and out 4 inches from the wall, it lessens any tendency to streaking of walls. Grilles must be fully closed or fully opened so as not to interfere with induction of room air or create noise. Grille surfaces utilize gaskets to prevent whistling when outlets are closed. Return inlets should have registers, but supply outlets can be regulated or throttled only by a damper in each supply duct in the furnace room. This allows the system to be balanced in the furnace room once and for all, not to be later upset by room occupants.

Furnace Unit

When the burner starts as the room thermostat calls for heat, air is quickly warmed. A switch responsive to temperature soon permits the fan to start; later, as the burner stops and the temperature drops too low, this same switch stops the fan. Quick circulation throughout the house soon stops falling room temperatures; shorter operation reduces operating cost. Fan speed is regulated by adjustment at the motor pulley, or by using a smaller pulley to give a temperature rise of 120F through the furnace. Final adjustment should have the furnace operating with a minimum on limit. Modulating control with gas is not yet available.

Return air may be arranged to enter the furnace left or right, Fig. 6, or through its bottom, Fig. 4, from where it passes over a filter to the fan, is heated, and is then discharged to the ducts after passing over a pan type humidifier. The filter may be removed by way of the front furnace panel. Furnace pit should preferably be waterproofed, or have a drain, since water elsewhere in the house may overflow, or ground water might enter at any crack. When the supply ducts are below the furnace, the warm air plenum may be inverted from that shown in Fig. 4 and 6, and extended down along the side of the furnace. This makes a better appearing and less-costly job, and reduces the pressure drop, due to the use of fewer elbows or no elbows at all, if the ducts in the floor are made to enter the side of the plenum.

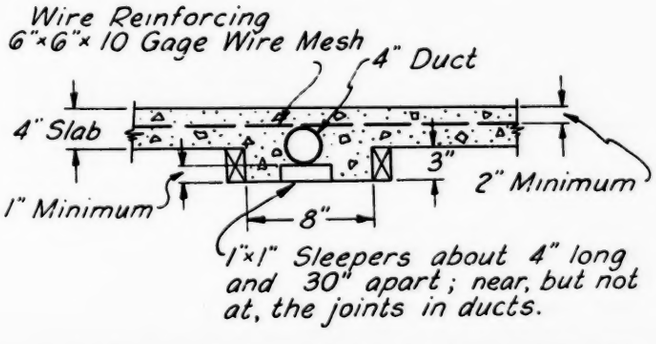


Fig. 3. Section A-A, Fig. 1, for ducts in first floor slabs.

To meet requirements of the American Gas Association and the Underwriters Laboratories, Inc., a

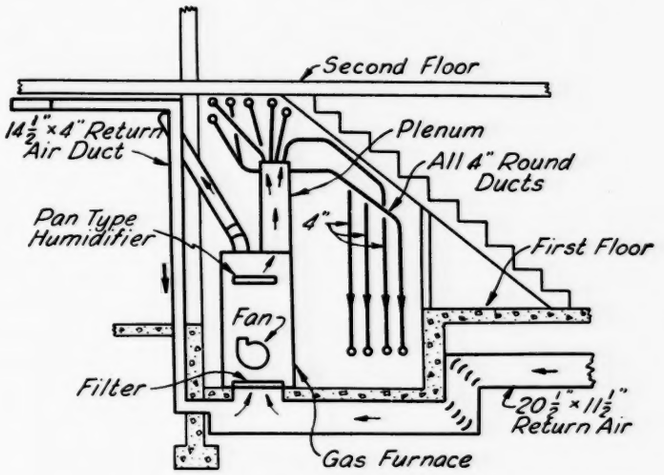


Fig. 4. Section through furnace room showing air supply and return.

decorative screen or louver that cannot be closed is required low down in the door of the furnace room, together with a louver high up in the door or above it. A third louver should be in the window. Each louver should have a free air area of one square inch per 1,000 Btu of furnace input.

The gas water heater should have a separate flue with opening to chimney higher than furnace smokepipe, so as to be favored by warm air from the larger device. Both flues should be galvanized and pitched up to chimney one inch in 10 ft. The chimney, by regulations, requires a clean out at the base. All smoke flues and warm air flues should be insulated and/or kept the prescribed distance from any combustible material. Incidentally, while Fig. 4 is the smallest furnace room the writer ever saw, it serves its purpose in every way.

Costs

The cost of an entire system to the general contractor has been in many cases only $600 to $700, of which slightly more than half is for the furnace plant. Of course, all costs are subject to the time, locality and number of houses involved, as well as the kind of fuel used. Gas is cheaper than oil for first installation cost. Not much can be saved by using, say, 4-inch galvanized rain water leader pipe in lieu of 4-inch aluminum ducts where the houses are few in number, see Table 1, but a development of 1,000 houses might mean a saving of $40,000. Savings can also be made by using fewer points of return air inlets, as possible in some house designs where the air can freely return of itself to a common inlet near the furnace. Even where numerous points of air return are required for more positive heating results, it may still be possible to economize by using a number of 7-inch round ducts of the required total area for the return air rather than a common duct of preformed asbestos. Savings

may also be made in some cases by insulating the ducts as described later under "Insulation."

Ducts of 4-inch round are of a convenient size for passing through construction, particularly when run in first floor concrete slab, and are most economical considering the added friction that would result from the use of smaller sizes. For example, a reduction to 3½-inch size would make only a negligible money saving and this at the expense of a friction loss more than doubled. In Fig. 1 and 2, the length of supply ducts totals 265 ft, of which 115 ft is in the first floor slab. In a building where the furnace is more centralized, any saving by size reduction would be still less noticeable, since the total linear length of duct then involved usually approximates but 200 ft.

Duct Materials

The cheapest materials from which ducts may be constructed is the first question that always arises. The answer is not difficult to find insofar as the cost of the material itself is concerned. However, some neglect to fully consider the cost in place and the results obtained. The principal objective should be to produce optimum performance with minimum cost, making all ducts smooth throughout with no projections and as direct as possible with a minimum number of elbows.

Turns at 45° should be used wherever possible. For example, in Fig. 2 the order of the three ducts to the bedrooms should be reversed, so that 45° ells can be used at the points of rise between the joints. The three ducts run as shown require 90° ells which could have been avoided by the suggested reversal. Turns in the first floor concrete slab should preferably be avoided, if possible, or removable slab placed over them. Degree of elbows should be made adjustable to meet conditions in the field.

Ducts of Transite, preformed asbestos and fiber have all received consideration, especially so due to their desirable values of lightness, insulation and sound reduction and their ability to withstand vermin and moisture. For cheaply built houses, the first two materials are usually too expensive, the third one does not offer sufficient economy. Concrete ducts have been used, but they require expensive form work, may or may not be smooth enough, and they lose considerable heat.

Fig. 5. Installation of air outlets near the baseboard avoids contact with drapes.

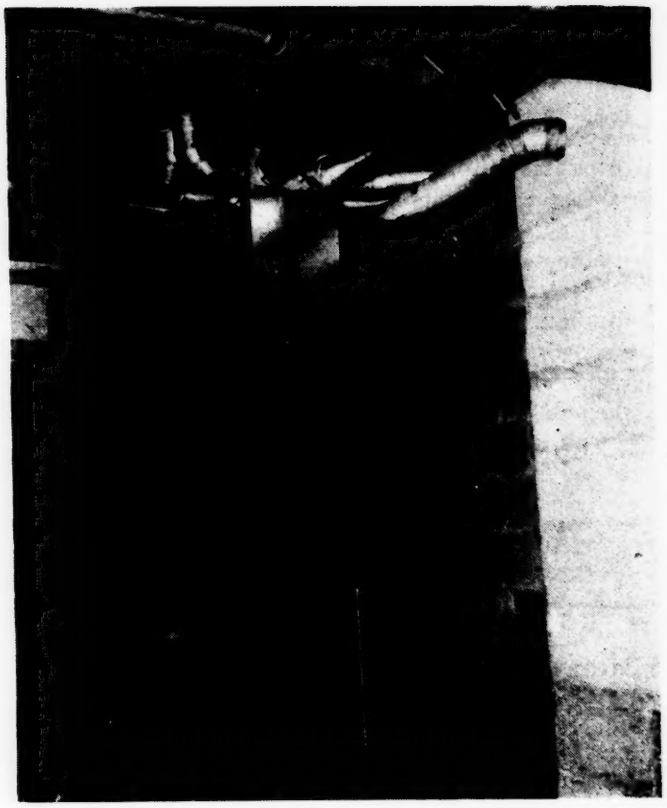


Fig. 6. Furnace installed with overhead plenum chamber.

Fiber conduit for electrical wires is highly resistant to water and electrolysis, has good mechanical characteristics and the joints are water-tight. On the other hand, heat and moisture might affect its smoothness in time and a vapor barrier may be required to avoid moisture pickup in summer months. The conduit has an odor which no doubt is more perceptible at higher air temperatures. Whether this odor would be sufficiently diluted in the air quantities handled is hard to say.

The use of round clay tile pipe increases pressure drop in their air stream, due to the additional joints, and also increases the heat loss. Tile with bell and spigot ends comes in two-foot lengths and is not cheap; plain end drain tile, which is cheap, comes in one-foot lengths. Six joints of tile if laid even, and there is no surety of this, give the same pressure drop as a single elbow. Tile pipe is subject to breakage during its transporting and handling and this results in an added cost for laying and for the cementing or taping of the numerous joints.

Galvanized iron ducts, or ordinary rain water leader pipe, should be plain and not the corrugated type. It is very cheap, particularly in the lighter gage, but it is subject to rust which in time may discolor walls at air outlets. This also results in unevenness and a decrease in cross-section due to rust accumulations. When such pipe is laid in concrete it is particularly important that the concrete mixture be fluid enough in accordance with best trade practice so as to flow

around the pipe, yet not flow into pipe joints. This method will provide a smoother bore and a proper support below the pipe should the pipe eventually disintegrate in spots or throughout.

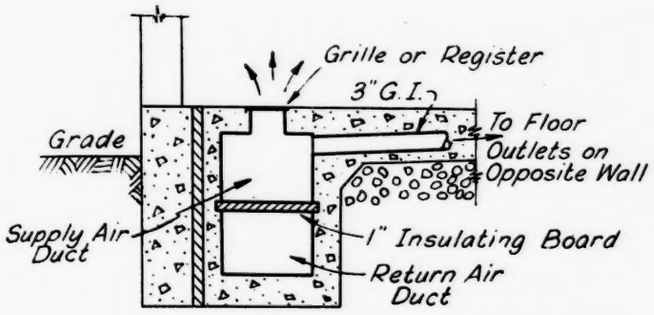


Fig. 7. Design combining supply and return ducts.

Aluminum ducts offer the preferred material since they are light, durable, very smooth, easily assembled with slip joints, and lose much less heat than one would normally think. Floor temperatures must be kept down, especially with an air temperature at the furnace bonnet of about 190F. Offhand, this temperature seems rather high as compared to radiant heating where the design is usually with a water temperature of 125F and the floor surface temperature cannot exceed 85F without discomfort. On the other hand, the warm air ducts are not as close together as are the pipes in hot water radiant heating systems. Furthermore, air to metal transmission is not as rapid as with water to metal.

One can buy 4-inch rubber hose that is substantial and rigid enough for the purpose in question at a cost of $20 for a 10-ft length. After the first floor concrete has hardened, the temporary hose form may be deflated and withdrawn for reuse. If enough houses are involved, substantial savings might be made. This method is being investigated by various builders and the economies depend upon how many floor slabs are poured in a day, and whether concrete is first poured only over the runs of hose as a protection when the entire floor is later poured, or whether the entire floor slab is poured at one time. Rubber hose has been successfully used in other work, such as in the removal of gas fumes through the ducts formed in the concrete slab of bus garages.

In industrial work, the roughness of any minute voids left at the interior surface of concrete pipe does not matter to such an extent as it might in small house work where the added friction to air flow could soon overcome the power of the fan. A further disadvantage is the additional labor required for forcing the concrete below and completely around the hose. Concrete must also be fluid enough for this purpose. In the final analysis, there is no surety as to the smoothness of a concrete duct and there is also a greater temperature drop or heat loss within the concrete duct, thus lowering its capacity for heat delivery.

Ducts in First Floor Slab

Ducts in first floor slab should be well removed from cinders. Asbestos paper tape should be pasted over each joint and the ducts should be painted with roof coating, asphalt, cement bituminous plastic thinned to brushing consistency, or similar protective coating.

The floor should be channeled where the ducts are to be located, Fig. 3, so that there is a minimum 2-inch concrete above the ducts to prevent unevenness of concrete and 1-inch of concrete on both sides and on the bottom. Wire reinforcing will probably be 6-inch square, No. 10-gage wire mesh laid above the ducts, and there should be a bed of gravel uniformly distributed 4 inches deep below the ducts. The channel is best made by using 1 x 1 inch sleepers 4 inches long on about 30-inch centers near, but not at joint centers. Where membrane waterproofing is desirable or required by local authorities, the membrane should be placed below the ducts in such a way that it will not interfere with the free flow of concrete around the ducts.

Some builders, instead of using building paper for the vapor seal, are using Sisalkraft paper, as this paper may be secured in various widths and in any length desired. For large housing projects, it may be desirable to get a roll the full width of the building. However, most builders advise that widths of 6 to 8 ft are about the limitations for one or two men to handle. Roofing felt is only 30 or 36 inches wide so that more seams are required. With Sisalkraft paper only two pieces could be used where seven or eight pieces would otherwise be required. Sisalkraft could be bought in 12-ft rolls for a 24-ft house and this would require only one seam.

Return Air Ducts

The minimum number of return air ducts are required to provide the necessary total or main return duct area. This area should be equal to and preferably

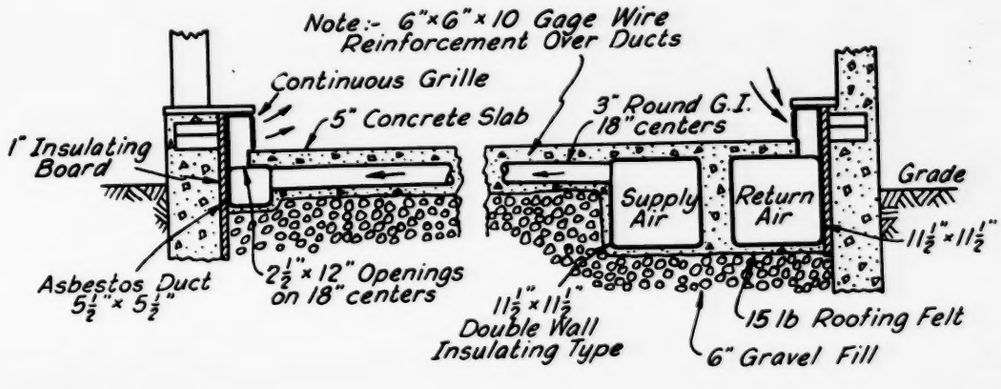


Fig. 8. First floor supply and return duct detail of design by Richard M. Stern.

greater than the combined area of the minimum number of supply outlets required.

Return air ducts in concrete must be strong enough to withstand the usual pourings. Double wall Careyduct (preformed asbestos) answers the purpose since it will not deflect and also has a good insulating value. Such ducts are painted with Carey Clad with branch connections sealed by an adhesive prepared by the same manufacturer. Rigid asbestos duct turns may be obtained. The duct comes with convenient slip joints. Such duct deadens sound, does not conduct electricity, and is vermin-proof. It is light and is capable of being sawed. When enclosed in concrete, it should have a waterproof membrane completely surrounding it for protection against concrete temporarily or permanently wet.

Duct of this kind can be used as a form in concrete for the upper concrete supply air duct, Fig. 7, thereby insulating the high temperature supply air duct against unusual temperature drop. The temperature of the supply air ducts must be kept up to keep down the number of supply ducts required, but in return air ducts, this is of no importance since the low temperature of return air can have but slight heat loss. As a rule, return air ducts have a much smaller total of heat losing area and, of course, any heat loss does not affect the number of returns required.

Another method of returning the air, and a cheaper one, is to use round aluminum ducts, a 7-inch duct for each 14 x 6-inch return air register, or where space permits, a 10-inch round from a 30 x 6-inch register.

Insulation

Unusually long runs of supply ducts should be insulated, otherwise they may have to be increased in size or number to give the required heat delivery. For example, an increase of 25% in duct capacity may be obtained by one inch of insulation on such duct runs. Again, suppose that the heat delivery capacity of a 4-inch round duct is only slightly below or above that of the heat loss of the space to be served. In such a case, the capacity may be made certain by insulating the duct and by taping its seams or joints.

Ducts which necessarily run in cold partitions, floors, walls or ceilings, should be insulated. Ducts carried up in warm inside partitions with horizontal

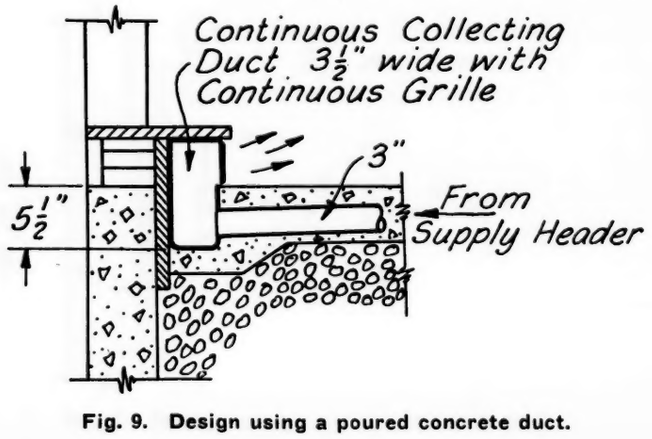


Fig. 9. Design using a poured concrete duct.

extensions on the second floor construction to exterior walls, not only avoid any need for insulation, but also give best distribution of the heated air. Ducts that run in joist space below attic floor insulation need no insulation, but ducts that necessarily run in an attic above floor insulation should be insulated with two-ply air cell asbestos or its thermal equivalent.

A crawl space below the first floor should be vented to outdoors during the summer season to prevent excessive condensation within the space. These vents should, of course, be closed off during the heating season. In crawl spaces, duct insulation should be vermin-proof, and of a kind which cannot be permanently damaged by moisture. Ducts, however, should preferably be run between the joists and above the floor insulation, in which case no duct insulation is required, provided, further, that the floor insulation is the thermal equivalent of two-ply air cell asbestos.

With ducts in first floor construction, with or without a crawl space below, the pleasant thought remains that there are no large pipes to contact your head, as most likely would be the case were there a basement in the house. Even when installed in a basement or crawl space, the pipes being small, take up no more headroom than do the lights. In any event, where they are laid directly in the floor, they are warming the floor for comfort of those above.

When ducts are laid in a first floor slab on the ground, the heat loss of the ducts should be estimated at 20% of the heat they carry in their air stream. A loss of only 10% may be assumed provided the exterior periphery of the house has 2 inches of insulation extending 14 inches below grade and provided, further, that a moisture membrane is used, together with one inch of insulation below the slab.

System Using Rain Water Leaders

Richard M. Stern, a consulting engineer of Seattle, Wash., has been working independently on a type of system similar to the one just described. By now several architects out his way who have used the method for small houses have found the heating both economical in first cost and operation and also quite satisfactory otherwise.

Mr. Stern uses a grid or header system of 3-inch (occasionally 2½-inch) galvanized rain water leaders, instead of carrying each duct back independently to the furnace. His outlets, while sometimes run continuous under a shelf, Fig. 8 and 9, are usually grilles or registers set in the floor along the walls, Fig. 7 and 12. Ducts for main supply air and for the collecting or sub-header warm air, Fig. 8 and 10, are all of ½-inch insulated preformed asbestos, but alternate schemes, Fig. 7 and 9, use poured concrete for the main ducts.

Mr. Stern advises that the use of concrete for warm air main ducts has usually been found more expensive than the ½-inch preformed asbestos duct. Furthermore, concrete has the disadvantage of greater heat absorption and heat loss. He adds, however, that the use of concrete for warm air ducts may be found satisfactory in small houses with runs not over 30 ft long,

and that concrete will in any event be satisfactory in the case of underfloor return ducts.

Return inlets are usually located in the baseboard. Return ducts are preferably run with sheet metal in the attic or above a hung ceiling in the corridor. In some cases, with small houses of five rooms or so, the return air may travel through a short hall direct to the furnace in the utility room, using no ductwork. A small outside air duct is run directly to the return air intake of the furnace to provide air sufficient for offsetting infiltration.

Main supply and return air ducts are placed alongside of one another, Fig. 8, in the concrete slab with supply grilles on the cold north wall. Upon occasion, the supply air duct is placed above the return duct, Fig. 7, or the return duct is omitted altogether, Fig. 10. The continuous grilles, Fig. 8 and 9, require a 4-inch shelf which need not necessarily take up usable floor space since tables and easy chairs can project over it. All of these are alternates and ingenious possibilities, according to the house, taste of the architect and owner, and the money available.

How the Grid System is Designed

The rain leader pipe or ducts are laid in the concrete slab of the first or ground floor, Fig. 8 and 10. They are ordinarily spaced as required, Fig. 11, to deliver a quantity of air which will transmit the room thermal load and duct heat loss at the temperature of air leaving the furnace. Usually the air quantities are 20 to 30 cfm per duct, which for 3-inch ducts, develop a total pressure loss of 0.05 inch of water when adding entrance loss to the duct friction. In other words, because of the added friction, long ducts must be more numerous (closer spaced) with each duct handling closer to 20 cfm. Short ducts would, therefore, be farther apart and deliver closer to 30 cfm. Temperature drop in the main warm air duct is assumed at 4F in 10 ft, when using insulated asbestos duct of only ½-inch thickness, but if this thickness calls for excessive air quantities, it should be increased to 1 inch, thus giving only half the assumed temperature drop.

By sizing the main supply air duct from the furnace at velocities graduated down from an initial 900 fpm, advantage is taken of some static regain to maintain somewhere near constant pressure throughout the main duct runs. The friction remaining is then only the static loss through the grid, which, of course, makes the system somewhat self balancing. Final balancing can then be done by means of a throat damper behind each outlet, or by blocking off the outlet collar through the concrete floor.

The maximum air temperature at the furnace bonnet with a grid system is usually held down to below 130F, often not over 120F. Under such conditions, the floor, when of bare concrete, is perceptibly warm and prob-

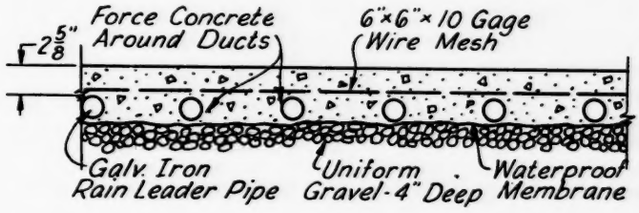


Fig. 11. Pipe spacing through section B-B of Fig. 10.

ably not over 80F. Readings of air temperature indicate a considerable portion of the total heat delivery to be by radiation from the warmed concrete floor panel.

Use of 2½- to 3-inch pipe, and therefore closer spacing, does produce a more uniformly radiant heated floor. Such sizes and spacing would produce a much greater heat loss to the ground than if 4-inch pipe were used were it not for the lower air temperatures within the pipes. There is, however, some additional heat loss, since more pipes are required and galvanized iron has a greater heat loss than does aluminum. Therefore, it is thought best that houses with a second floor should have a separate duct from the furnace, rather than to attempt to take risers off the ground floor asbestos ducts. This provides a conventional type warm air system for the upper floor. A number of houses have been so planned, the duct work dividing at the furnace bonnet to form a split system. A separate duct is provided for each floor and each duct is balanced or zoned by the operation of throttling dampers.

Grid System Cost

The cost has been $1,000 and upwards for a house with an area of about 900 sq ft. It appears as if costs will average about $1 per sq ft. Mr. Stern feels that his forced air split system for small houses of about 900 sq ft area are probably about 20 to 30% lower in first cost than the usual hot water floor radiant installation. As for larger houses, costs for forced air and radiant hot water are about the same. In any event, his particular rain leader design was not developed primarily for low cost heating, but rather for floor warming in residences having too high a heat loss for a floor panel job. He further states that there was

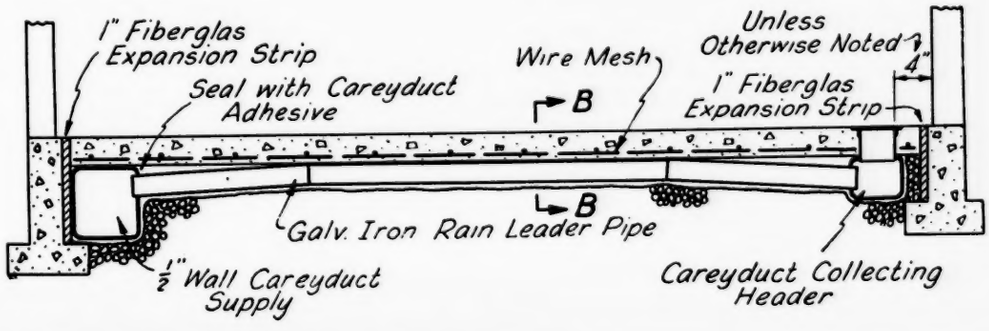


Fig. 10. Cross section of first floor with supply only.

also a desire to provide the greater advantages of forced air over that of ordinary hot water.

Locating the Air Outlets

Mr. Stern favors locating warm air outlets in the floor, Fig. 12, where they are usually set about 4 inches

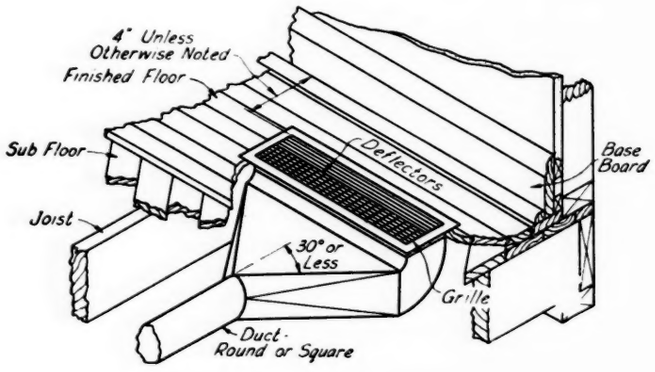


Fig. 12. Details of flocr installation.

out from the finished wall under the larger windows. Such location, together with a 10° air deflection from the wall avoids fluttering of the drapes and also forces the air to cover more window area. Incidentally, this floor setting is ideal for use with summer cooling.

Floor outlets as used are modern, quite sanitary, and usually small in size. They are 4-inch wide by 10- to 14-inch long. Multi-louver shut-off is added only in bedrooms. The usual walnut brown electro-plated finish is pleasingly unobtrusive in appearance to most owners. However, outlets can be finished to match interior decorations. If desired, outlets can be rabetted into the floor. While furniture such as a chair or davenport may stand almost over an outlet, it may be said that furniture seldom blocks off floor outlets as sometimes happens with baseboard registers.

Warm air registers in bath rooms and kitchens are installed 6½ ft from the floor, although occasional baseboard placement is used in the kitchen. In addition, about 60 to 80 cfm of the kitchen supply is introduced through the floor of the towel drier compartment with the overhang of the toe space cut out for relief. Cold air returns, of course, should be located in the baseboard regardless of the room.

How Comfort is Obtained

Since air diffusion is jetted upward along outside walls or major windows where cold air is offset before it travels but a matter of inches, it follows that the occupied space is free from drafts and that condensation on windows is reduced to a minimum. Floor to ceiling temperature differentials are also reduced with consequent comfort and fuel economy.

Since there are no drafts blowing across ankles as with some baseboards registers, the fan thermostat can be set for almost continuous fan operation with 110F on and 80F off, rather than with the higher settings often employed in conventional systems. The low setting permits use of high wall registers in kitchen and bathroom where floor registers cannot be used.

With night set back of room thermostat, the fan stays off most of the night, but during the day in weather below 50F, the fan operates almost continuously. Semi-continuous fan operation prevents dust or dirt from settling into the floor outlets during the heating season, while the low supply air temperature is economical and prevents streaking of walls.

Conclusion

When the client has but little money the architect designs him a small house—the most house for the money. As a rule, in such cases, there is no money left for even a basement, so by the time the architect turns his thought to the heating plant, one is tempted to install a simple pot bellied stove in the living room and let it go at that. In any event, before building a house without a basement, one should first check the local code since some town codes do not allow such an unorthodox practice; for example, certain town codes in New Jersey are not in accord with such an omission, while others, like Morristown, N. J., are still working it out by means of a new code.

Leak Detector Speeds Control Inspection

At the Thermostat Division of Robertshaw-Fulton Controls Company, Bridgeport, Conn., a General Electric Type H Leak Detector has helped speed production and improve product quality.

The instrument is used to test for leaks in experimental thermostats and certain types of finished units. Before it was installed, the units were tested by introducing air at pressures up to 50 lb and submerging them in water, escaping air bubbles indicating a defect.

Now a halogen gas, Freon, is introduced as a tracer into the closed thermostatic systems to be tested, after which the operator passes the nozzle of the detector over the soldered joints. Use of this testing procedure, it is claimed, enables the company to locate leaks in about one-fourth the time it took with the old method.

A company technician said: "The leak detector saves time in both laboratory and production-line testing, because now we don't have to bother with the bulky tanks, fixtures, and other pieces of special equipment formerly required. With the old method each type of thermostat required a different sort of leak test." He stated that from an operator's standpoint the detector is easier to use than the old method for the operator can now sit down and work faster.

What Happened at Minneapolis

Summary of papers presented at the semi-annual meeting of the American Society of Heating and Ventilating Engineers held at Minneapolis, Minn.

THREE technical sessions were arranged for the semi-annual meeting of the American Society of Heating and Ventilating Engineers that was held at the Nicollet Hotel, Minneapolis, Minn., June 20-22. Presiding at these sessions were Alfred E. Stacey, Jr., president; Lester T. Avery, first vice president; and L. E. Seeley, second vice president. In addition, the program included a number of special entertainment features for the ladies and children; a banquet on Wednesday evening at which Bob DeHaven, master of ceremonies for several local CBS radio station programs served as master of ceremonies, and a golf tournament. Registration was over 400 persons.

The Minnesota chapter was host to the society, with John E. Haines serving as general chairman of the committee on arrangements, L. C. Gross, vice chairman, and F. B. Rowley, honorary chairman. J. S. Locke headed the entertainment committee.

Following are brief abstracts of the papers presented at the technical sessions.

Blower Performance

A test arrangement and procedure to determine the performance of blowers used in conjunction with various forced warm air furnaces having a maximum output of 150,000 Btu per hr were described in a paper, "Determining Performance of a Blower Used With a Furnace." The authors are N. A. Buckley, special research assistant and R. W. Roose, special research associate, University of Illinois. The studies and tests indicate:

1. Blowers of the same type and nominal size, but produced by different manufacturers, do not necessarily perform identically when operated with the same furnace.

2. It is advisable to use a substitute for the furnace filters when conducting furnace performance tests. A set of filter substitute plates have been developed for this purpose.

3. Variable resistances can be applied for the furnace filters when conducting performance tests on either the inlet or the discharge of the furnace without changing the performance characteristics of the furnace-blower combination.

4. Increase of resistance due to heating of the air by the furnace can be calculated. Over the narrow range of temperature rises encountered in furnace testing, the pressure drop through a blower-furnace combination will be 8% greater when operated with heat input than when operated with no heat added. Pressure drop with heat added = (Pressure drop with no heat added) $\times$ 1.08. Therefore, except for establishing rated operating conditions, conducting tests with heat added is not considered essential.

The paper described the equipment tested and results obtained with various blowers used in the tests.

Perforated Panels

A study was made of air streams discharged from perforated ceiling panels with special reference to the effect of such variables on the gross area of the panel, the size and spacing of the perforations, and the height of the ceiling. This was covered in a paper on "Air Streams from Perforated Panels," by Alfred Koestel, instructor; Philip Hermann, instructor; G. L. Tuve, professor—all of the Department of Mechanical Engineering, Case Institute of Technology, Cleveland, Ohio.

A method was made available for estimating the performance of perforated panels within the range of conditions covered by the tests—3 to 40% free area, 1 to 14 sq ft panel size and up to 3,500 fpm air velocity through the perforations. Moderate variations in panel shape or the hole spacing had no apparent effect on the performance.

In the case where low residual velocities are desired near the panel discharge, for large volumes of primary air supply, the optimum panel design and arrangement would be obtained if panels of small area and low per cent free area were spaced in such a way as to prevent the streams of each outlet from converging and forming higher stream velocities. With such small panel areas the phase lengths are decreased and with low free areas, the diffusion is increased.

In addition to the perforated metal panels that were tested, studies were also made of perforated transite panels.

Box Plenum

"Pressure Loss and Air Flow Characteristics of a Box Plenum," were described in a paper by S. F. Gilman, R. J. Martin, W. R. Hedrick and S. Konzo. The authors are, respectively, special research assistants, and professor of mechanical engineering, University of Illinois. A study was made of a box plenum with six branch outlets. Results obtained in the tests include the distribution of air to the outlets, the flow pattern inside the plenum, and the outside pressure losses. Five separate entrance sections for introducing air into the plenum were investigated. The investigators, as the result of this study, arrived at the following conclusions:

1. The total pressure loss associated with a box plenum is composed of an entrance, turbulence and outlet loss. The entrance and turbulent losses can not be logically separated.

2. The pressure loss at the entrance to the branch outlets is in good agreement with the generally accepted value.

3. The pressure losses and air flow conditions within a box plenum are not affected by the addition of equal resistance to each outlet.

4. For all practical purposes, the losses with the

remote and close coupled fan positions are the same. Therefore, the results will apply to field conditions when the fan is located very close to the plenum.

5. Take-off fittings for branch outlets should not be located within 12-inches of the upstream corners of the plenum if excessive balancing losses are to be avoided.

6. The plenum loss with the abrupt expansion entrance section (conventional type) installed is much higher than the calculated entrance loss alone, indicating the loss due to turbulent mixing in the plenum is severe.

7. Unless the entrance section of a box plenum is carefully designed, large pressure losses and unstable distribution of air to the branch outlets will result.

8. When the conventional type, diverging section, and an internal expansion ring with five vanes inside the plenum entrance sections were used, rotational flow occurred and the system was unstable. The use of these three types should be avoided.

9. The entrance section of a type with the internal expansion ring modified at the upstream end and placed inside the plenum, is superior to the other types tested from the standpoint of total loss, pressure required to balance the system, stability of the system, and space requirements. This section is recommended for use in the field. The plenum loss with this section can be taken as 0.90 velocity heads in the inlet duct, provided no branch take-off is located within 12-inches of the upstream corners of the plenum.

Earth as a Heat Source or Storage

"Earth as a Heat Source or Storage Medium for the Heat Pump," was covered in a paper by E. W. Guernsey, P. L. Betz and N. H. Skau. Respectively, the authors are director of research, assistant director of research, and research chemist, Consolidated Gas Electric Light and Power Co. of Baltimore, Md.

For the sample calculations that are used in the paper, the authors consider the case of a small, well-insulated house in an area where the minimum temperature is 0F, the temperature deficiency is 4,500 degree days per season, and the total heating requirement is 75 million Btu. It is assumed that the coefficient of performance for the heat pump is 3, and that 50 million Btu of heat are extracted from the earth annually. Surfaces are assumed to be placed in a soil of 25 to 30% moisture.

A consideration of shallow coils at various depths suggests that not less than 6,000 to 7,000 sq ft of overall coil extent would be required to supply the heat requirements of the small house considered, if the minimum temperature at the evaporator is to be not lower than 15 to 20F. The required surface would drop rapidly with a reduction in minimum permissible temperature at the evaporator.

When the heat transfer surface is in the form of a plate or planar coil, the surface required is only one-fourth as great for balanced heat extraction and return as for heat withdrawal only.

Since the extent of coil needed for the various cases considered in the paper is quite large from the standpoint of installation cost, the authors doubt that ground coils will be an economic heat source for general use.

Heat Pump Operation

"Heat Pump Results in Equitable Building," were described by J. Donald Krocker, consulting engineer; Ray C. Chewning, office of J. Donald Krocker; and Charles E. Graham, building engineer, Equitable Building, Portland, Ore.

This system, which has been operating for a complete year, is considered, at present, the largest heat pump installation in an office building in the United States. Operating results are included in the paper and the system was described.

A number of general observations made by the authors may be of value to those concerned with the design or operation of a heat pump in a large building.

1. The heat source normally adopted will probably be well or surface water. Often such water is corrosive or may form accretive substances on metals. Water analyses should be made in all cases to determine required treatment, if any.

2. Feasibility of heat recovery with the heat pump systems due to the quantities of water and air being handled is definitely substantiated by experience of operation on the Equitable Building, as is the economy of using the waste water for pre-conditioning incoming ventilation air.

3. The importance of applying the best possible automatic control in the interest of operating economy cannot be over-emphasized. Because it is necessary to use small temperature differentials in the supply and return water and in the supply air, the control must be accurate and it must be flexible. Flexibility is

Seated at the speakers' table at the banquet held on Wednesday evening are, left to right, Frank B. Rowley, honorary chairman, committee of arrangements; Mrs. Richard C. Jordan; Ernest Szekely, treasurer; Mrs. L. C. Gross; Lester C. Avery, first vice president; Mrs. Alfred E. Stacey, Jr.; John E. Haines, general chairman; Bob DeHaven, master of ceremonies; Alfred E. Stacey, Jr., president; Lyman C. Gross, vice chairman, committee of arrangements; Lauren E. Seeley, second vice president; Mrs. F. B. Rowley; Richard C. Jordan, president, Minnesota chapter.

required in order to maintain the most efficient conditions of operation for the refrigeration equipment. For example, while a coefficient of performance of 5 or 6 may be obtained on a basic refrigeration cycle with a condensing temperature of about 80 to 85F, the coefficient may drop to as low as 3 when this temperature is raised to 105 or 110F.

4. In providing for the automatic change of water circuits from one type of operation to another, care must be exercised in arranging the opening and closing of valves in sequence to prevent damage from water shock or hammer.

5. Sufficient recording instruments of various types should be installed. Such instruments include flow meters, temperature recorders, room temperature and humidity recorders, together with units to control the outside air fans and exhaust fans, etc.

6. The operator must have an analytical mind which can sort out and properly correlate the information which the recorders give.

Ground Temperatures

A. B. Algren, professor and head of the division of heating, ventilating and air conditioning, mechanical engineering department, University of Minnesota, told of studies made of "Ground Temperatures as Affected by Weather Conditions." A 110-acre plot was used as a test area over a two-year period and thermocouples were employed to measure soil temperatures to a depth of 16 ft.

Extensive tables were presented to show ground temperatures for both bare ground and for ground covered with sod at various depths up to and including 16 ft for the period from August, 1947 to February, 1948. Due to a time lag, it was found that at 16 ft, the ground is warmest in November and coldest in April.

In connection with this study, tests were also made of the thermal conductivity of the soil. Results indicate that the conductivity of the silt and clay soils varies as follows:

1. At a constant moisture content, it increases with an increase in dry density. The rate of increase is fairly constant and is independent of the moisture content.

2. At a constant dry density, it increases with an increase in moisture content.

3. For the unfrozen soil, it increases with an increase in mean temperature.

4. The frozen soil at low moisture content shows very little change; in general, for soils of high moisture content it shows an increase for a decrease in temperature.

5. When changing from unfrozen to frozen soil, the conductivity varies according to the moisture content. For dry soils, it does not change; for soils of low moisture content, it decreases; and with soils of high moisture content, it increases.

Measuring Permeance

An apparatus for measuring the water vapor transmission of materials was described in a paper "Permeance Measurement Improved by a Special Cell." The paper was prepared by F. A. Joy, associate professor of engineering research, and E. R. Queer, professor of engineering research, The Pennsylvania State College, State College, Pa. The apparatus can accommodate not only paper-thin material but thick specimens as well.

Four different models were constructed. In the last design, a duplex cell is used with the test specimen ring mounted vertically between the two sections. The water and desiccant containers are of special design involving primarily a vapor tight can with two openings through which air may be drawn by a fan when a valve in the entrance port at the top is opened. A feature of this cell is positive air movement over the specimen surface. While the air velocity is variable, it approaches a maximum of 350 fpm. The design fulfills the requirements for a vapor transfer of 19 grains per hour, the maximum tested thus far.

Mean test temperatures have ranged from 50F to 110F with a temperature difference as high as 23F between panel surfaces. Relative humidities on the water side were set and controlled at values ranging from 30 to 93%. Operation on the desiccant side included humidity values from 15 to 77%. Tested vapor transfer rates included values from 0.06 to 19 grains per hour. Tested specimens ranged in permeance from 0.34 to 120 Perms.

While Model 3 and Model 4 have important uses, Model 3 is simpler and requires only a temperature controlled room for its operation. It is considered best for testing materials of lower permeance which are known to be insensitive to relative humidity. Model 4 is limited only by its heat transfer capacity.

Human Tolerance to Extreme Heat

Results of 21 human exposures to air temperatures ranging from 140 to 240F were presented in a paper on "Human Tolerance Limits for Extreme Heat," by W. V. Blockley, research associate, and Craig L. Taylor, associate professor, department of engineering, University of California. A summary of the findings of the tests follow:

1. Skin temperature rises continuously during exposure, the rate of rise being successively greater with increasing ambient temperature levels. The maximum mean skin temperature observed was 107F, reached in an exposure at 240F.

2. Heart rate increases in a similar fashion to skin temperature, usually reaching values between 140 and 160 beats per minute.

3. Blood pressure changes show greater individual differences than do other physiological factors. Systolic pressure rose during exposure in all cases, while diastolic pressure declined continuously in one subject, but remained relatively stable or slightly elevated in another.

4. Electrocardiograms taken during and after heat exposure show minor variations from the normal resting records of the individual, but no distinct indications of heart damage have been seen.

5. Exhaled air temperature responds rapidly to the

heat load, and appears to be a useful indication of thermal stress. Measurement of temperatures at several locations in the respiratory passages indicates that the mucous membranes exert a strong regenerator influence, cooling the inhaled air by as much as 100F in a few inches of travel, and warming the exhaled air, though to a much lesser degree. However, as far as can be determined, because of the evaporative loss the total respiratory system continues to serve as an avenue of heat loss even under conditions close to the extremes of these studies.

6. Respiratory ventilation increases markedly under heat stress, often reaching values two or three times the resting levels, though respiration rate is not significantly affected. Oxygen consumption apparently follows a course that is roughly parallel to that of ventilation.

7. Subjective symptoms of thermal strain are numerous, but only a relatively few were observed consistently in each experiment. One common feature of all exposures carried to tolerance is a feeling of air hunger, associated with deep and irregular respiration. This symptom may be accompanied by restlessness and nervous irritability, while waves of dizziness may intervene at terminal stages.

Cooling Loads

C. O. Mackey, professor of heat power engineering, and N. R. Gay, associate professor of heat power engineering, Cornell University, prepared a paper read by Prof. Gay on "Heat Gains Are Not Cooling Loads." There are so many variables that influence the true cooling load in addition to the energy release rates that finding the true cooling load does not lend itself to an exact analytical approach. A number of examples were given of the relation of these two factors, and from these examples, tentative general conclusions were drawn.

In cases where there are steady sources of energy emission, the instantaneous contributions to the cooling loads from steady sources of energy emission having appreciable components of radiation, may be significantly less than the instantaneous rates of energy emission.

In cases where there are periodic sources of energy emission:

1. The instantaneous contributions to the cooling load from sources of periodic energy emission like roofs, walls and sunlit glass may be less than the instantaneous rates of energy emission at the time of maximum cooling load.

2. The instantaneous cooling loads may be appreciably less than the instantaneous rates of energy emission under the following conditions: When precooling or after-cooling is used; when a constant air temperature is maintained; and when any appreciable portion of the structure is in heat exchange relationship with a relatively cool heat sink, like the ground, in addition to being in heat exchange relationship with the sun and outdoor air.

3. If a steady operative temperature is maintained by lowering the temperature of the air as the mean radiant temperature of the enclosure rises, the instantaneous maximum cooling load may not be significantly less than the instantaneous rate of energy emission at the time of maximum load.

Solar Energy Through Glass Block

George V. Parmelee and Warren W. Aubele of the ASHVE Research Laboratory, Cleveland, Ohio, presented a paper on "Solar Energy Transmittance of Eight-Inch Hollow Glass Block." It covered five patterns of hollow glass blocks. The authors summarized the results of the investigation as follows:

1. The shading effect of the mortar joints is a principal factor in establishing the solar energy transmitting characteristics of glass block panels, except in the case of prism-type blocks. Increasing obscurity of pattern design and use of screens reduce energy transmitttance proportionately.

2. The light directing characteristics of the prism-type blocks cause their transmitting characteristics to depart considerably from the trend indicated by mortar joint shading.

3. Heat gain data due to absorbed solar energy must be recognized as approximations that are subject to considerable variation and therefore must be used with caution.

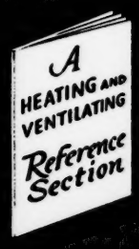
4. If design boundary conditions of heat transfer had existed during the tests, values of overall coefficients for glass block panels would be somewhat higher than those presented. The possible influence of boundary conditions on panel conductance needs further study.

Ventilation for a College Library

The new library annex of the University of California, in Berkeley, will be served by fans delivering 92,000 cfm. Just completed at the Berkeley plant of Westinghouse Electric Corporation's Sturtevant Division, the fans are designed to provide the required amount of fresh air with the least possible sound—a particularly important consideration when designing a ventilating system for a library.

The wheel with backwardly inclined blades will turn at the relatively slow speed of 250 rpm. The 15 hp Westinghouse motor powering each fan is of the silent type, turning at 900 rpm and transmitting its power to the fan shaft through a noiseless V-belt reduction drive.

The project was designed by Arthur Brown, Jr., architect, and Thomas B. Hunter, mechanical engineer. Installation is being done by the Ace Sheet Metal Works, of San Francisco.



Calculating Heat Gains in Air Conditioning Loads

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The heat gain calculation procedure described has been used by one manufacturer in competitive practice for several years. It has been thoroughly field tested and has been used to determine the cooling load for equipment selection.

BEFORE air conditioning installations can be made on a scientific basis, it is necessary to know not only what kind of equipment to use, but also how large the equipment should be.

The determination of size of equipment to use is reached in two steps:

(1) A heat gain calculation is made to determine what cooling capacity is needed to do a satisfactory air conditioning job.

(2) Reference is made to test data to find what sizes of air conditioning units and condensing units are needed to give the required cooling capacity.

This article concerns itself with the theory and mechanics of the first step—the heat gain calculation.

Heat Gain of a Structure

In warm weather there are, in general, five different sources from which heat flows into a structure. Each of these sources is important and must be taken into account: Leakage of heat through walls, windows, etc.; heat brought in with outdoor air; heat liberated by people; heat generated by lights and appliances; heat radiated by the sun.

These sources of heat are now fairly well understood and are susceptible to calculation which may be quite accurate.

The sun is the final source of heat gain which needs to be considered here. Heat leakage into top floor rooms may be more than doubled because of this effect. Glass windows exposed to the sun may allow as much as 90 per cent of the incident heat to pass through.

The making of a complete heat gain calculation involves not only the independent calculation of each of the five components, but it involves also the proper addition of these components to determine the correct maximum heat gain for the structure. The underlying principles are fundamentally simple, and, in fact, the whole process of calculating each component and adding it in proper relationship to the others has been so organized and simplified that it may be carried out with little more effort than filling out a standard 8½ inch by 11-inch calculation form.

Heat gain is composed of two factors. One is called the *sensible heat gain*, because it is associated with a change in temperature of the air, and may be sensed by that temperature change. The other is called the *latent heat gain*, because it is unaccompanied by any temperature changes and involves simply an increase of the moisture content of the air. In this sense it is latent, or hidden. Since the elimination of excessive moisture by an air conditoning system requires the removal of heat from the air by the cooling equipment, this moisture gain represents a gain of heat.

The several components of the cooling load are classified in this manner as follows:

(1) Sensible heat only.
   a. Conduction through walls, etc., (temperature difference).
   b. Sun effect on walls and roofs (additional conduction).
   c. Sun effect through windows.
   d. All heat generating appliances which do not give off moisture.

(2) Both sensible and latent heat.
   a. Ventilation air.
   b. Occupants.
   c. Appliances which give off moisture.

Design Problem

In what follows, the various components of the cooling load will be discussed individually, and then consideration will be given to the manner of adding them. To clarify discussion, the following example is used:

It is proposed to cool an office 20 ft × 15 ft with 12 ft ceiling, one exposed wall, facing south, in which are 50 sq ft of window area, and located in 40 deg north latitude zone. The following data apply: Outside wall, (12 inches brick) $U_w = 0.36$; inside walls $U_w = 0.27$; floor and ceiling, $U_w = 0.32$; window area, 50 sq ft; occupants, 3; moderate amount of smoking; lights, two 100-watt lights used in daytime.

Calculation of Heat Leakage Through Walls

The technique of calculating heat leakage through a wall was first explained some 300 years ago by Sir Isaac Newton, and by Fourier, the French mathematical wizard. They showed that heat conduction was dependent on three factors:

(1) The area of the wall.

(2) The heat transmissive quality of the wall.

(3) The difference in temperature between the air on the two sides of the wall.

The precise relationship may be given by the formula: Heat gain in Btu per hr =

(wall area) $\times$ U_w $\times$ (temp. diff.).

In this formula the factor U_w is known as the overall heat transfer coefficient of the wall, and is usually expressed in Btu per hr (sq ft) (degree temperature difference). The wall area is expressed in square feet, and the temperature difference in degrees Fahrenheit. The precise means of determining each of these factors will be discussed. The product of all three factors gives the heat gain.

Determining Wall and Glass Areas

Wall and glass areas may be determined in the usual manner from the linear dimensions, which are ascertained originally in making the survey, or else taken from available blueprints. For the purpose of this calculation it is necessary to know the net wall area, in order that the heat flow through the wall proper and through the windows may be calculated separately. The net wall area is simply the difference between the gross wall area and the window area. Areas of ceilings and floors are determined in similar fashion, skylights being treated simply as windows in ceilings. In the case of a sloping roof, use the horizontal projected area of the roof. The effect of the slope is taken into account in the heat transfer coefficient. See Table 1.

Determining Heat Transfer Coefficients

The overall heat transfer coefficient of a wall is a measure of its ability to conduct heat from the air on one side to the air on the other. This heat transmissive ability depends on three things:

(1) The air velocity on the outside of the wall.
(2) The air velocity on the inside of the wall.
(3) The thermal properties of the wall.

Considerable research on this subject has been done in this and other countries, and it is now possible to find directly the overall heat transfer coefficient, U_w, for any of the usual wall, partition, roof, floor, or window constructions, simply by referring to standard tables. The coefficients are usually based on an assumed wind velocity of 15 miles per hour across outside surfaces, and still air conditions (15 to 25 fpm) indoors.

Certain special cases, not covered by the tables directly, arise in connection with rooms under attics and sloping roofs. Table 1 shows how coefficients for such special cases may be derived. If coefficients are determined in this way, then one may ignore the slope of pitched roofs and calculate the areas as though the roofs were flat. Table 1 also shows how to determine temperature differences and sun effect coefficients for these special cases.

Determining Temperature Differences

It will be evident that to arrive at the temperature differences across an outside wall one must first settle on an outdoor temperature to be used as a basis for design, and then decide what indoor temperature it is desired to maintain.

What outdoor temperature should be used as a basis for design? A choice of this temperature requires a comprehensive understanding and correct interpretation of weather bureau data. The highest outdoor temperature ever recorded anywhere in the world was near Death Valley, Calif., where 134F was recorded on a properly shielded and tested thermometer. Yet in San Francisco, only 500 miles away, the mean dry bulb temperature for July is 58.5F, although the maximum ever recorded there for this month was 99F. In Phoenix, Arizona, the mean dry bulb temperature for July is 90.5F; in New York City it is only 73.5F. And there were places in the Middle West in the summer of 1934 where for over 30 consecutives days the temperature never failed to pass 100F.

These wide variations in temperature indicate that considerable care must be exercised in the selection of the outdoor design temperature. It is not reasonable to choose the highest temperature ever recorded in a given locality, for that may occur only once in 50 years, and then for only a short

TABLE 1—TRANSMISSION COEFFICIENTS, TEMPERATURE DIFFERENCES AND SOLAR RADIATION COEFFICIENTS FOR CEILINGS OF VARIOUS ROOMS

Location of Room	U_w	Temperature Difference	R_w
Not on top floor	$U_w = U_c$	$(t_o - t_i)$ — (3 to 5 Deg.)	$R_w = 0$
Under unventilated attic —with sloping roofs— ordinary house construction	$U_w = \dfrac{U_c}{1 + 0.7\dfrac{U_c}{U_r}}$	$(t_o - t_i)$	$R_w = F \times a \times I$
Under unventilated attic —with flat roof	$U_w = \dfrac{U_c}{1 + \dfrac{U_c}{U_r}}$	$(t_o - t_i)$	$R_w = F \times a \times I$
Under well ventilated attic —natural ventilation	$U_w = U_c$	(Attic temp. $- t_i$)	$R_w = 0$
Under fan ventilated attic	$U_w = U_c$	$(t_o - t_i)$ + (5 to 10 Deg.)	$R_w = 0$
Directly under flat roof— roof also forms the ceiling	$U_w = U_r$	$(t_o - t_i)$	$R_w = F \times a \times I$
Directly under sloping roof —room also forms the ceiling	$U_w = \dfrac{\text{Roof area}}{\text{Horizontal} \times U_r \times \text{Ceiling Area}}$	$(t_o - t_i)$	$R_w = F \times a \times I$

LETTER SYMBOLS

t_o = design outdoor dry bulb temperature.
t_i = design indoor dry bulb temperature.
U_w = overall transmission coefficient.
U_c = transmission coefficient for ceiling alone.
U_r = transmission coefficient for roof alone.
a = absorption coefficient. See Table 8.
R_w = solar radiation coefficient.
F = radiation factor. Determine F to use in fourth column from formula on page 92 by using the value of U_w calculated in the second column.
I = intensity of sun. Determine I to use in fourth column from Table 7, using values for horizontal surfaces.

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time. Equipment chosen on such a basis would be much larger than it need be. Likewise, it is not reasonable to choose the average summer temperature, because there are many days in summer above the average temperature. Equipment chosen on this basis would be too small for the job.

In general, the design temperature should be high enough so that it will include about 90 per cent of the daily maximum temperatures during the cooling season. Then there will be a few days on which the equipment installed will be somewhat inadequate during a few hours of the day, but it will be generally satisfactory and will not be unduly expensive.

Table 2 gives outdoor dry bulb and wet bulb temperatures that may be used as a basis for design in a few cities of the country. Similar data are available for most of the major cities.

The desired indoor design dry bulb and wet bulb temperatures and absolute humidity may be determined from Table 3. In addition to the data given, in some localities it is competitive practice to design for 8-10F difference between indoor and outdoor dry bulb temperatures when the customer occupancy time is 15 minutes or less. A good example of this type of application is a drug store. It is common practice to design for an indoor dry bulb temperature of 78 to 80F and 50 per cent relative humidity for an occupancy of one hour or more.

These data are complete enough to permit arriving at design temperature differences for outside walls. Design temperature differences for ceilings of rooms under attics, or under flat or sloping roofs, are given in Table 1. For inside walls or partitions, where there are no unusual sources of heat on the far side of the partition, it is justifiable to assume a temperature difference from 3 to 5F less than for the outside walls. Where a partition leads to a hot kitchen or similar heated space, it is desirable to assume a temperature difference somewhat greater than for the outside walls.

It should be noted here that ordinarily the outdoor temperature is highest at about 3 p.m. Thus by the methods described, one automatically makes the calculation for this time of day. There are many cases, however, where the conditions of occupancy, sun effect, etc., are such that the maximum total heat gain from all sources occurs at some other time of day. The overall calculation should be made, of course, in such fashion that it gives the greatest total heat gain occuring over a 24-hour period. It is desirable, therefore, to determine the time at which this greatest heat gain occurs, and correct the value of the conduction component from 3 p.m. to the proper time. The precise manner of making this correction will be explained later.

TABLE 2—SUMMER DESIGN TEMPERATURE

City	Dry Bulb Temperature	Wet Bulb Temperature
Birmingham, Ala.	97	78
Cleveland, Ohio	92	74
Detroit, Mich.	97	75
New York, N. Y.	92	76
San Francisco, Calif.	89	64
St. Louis, Mo.	101	77

TABLE 3—DESIGN INDOOR CONDITIONS FOR SUMMER

Class of Application	Dry Bulb, F	Wet Bulb, F	Abs. Humidity, Gr. per lb.
High class comfort cooling where maximum comfort is of prime importance and where average of occupancy is over 1 hour	78[1]	67	82
General class of application in which it is desired to maintain comfortable conditions for periods of occupancy of 1 hour or more but where cost is an important consideration, and partial cooling will be acceptable on a greater number of warm days than for application above	80[2]	68	84
Applications in which the period of occupancy is less than 1 hour	82	69	86

[1]It will seldom be necessary to provide more than 20F design temperature reduction.

[2]In no case is it desirable to provide a design dry bulb temperature reduction of less than 8F.

The importance of selecting the correct design conditions for the particular job cannot be overestimated. A careful decision as to the amount of temperature reduction to be provided will determine to a large extent the degree of satisfaction which the installation will give.

Heat Leakage Problem

As an example of proper procedure, the heat leakage for the office described under "Design Problem" is now calculated. Assume a design outdoor temperature of 96F and the design indoor temperature for an occupancy of more than one hour is 80F. The design temperature difference is, therefore, 16F.

Outside wall

Heat leakage, Btu per hr $=$ $U_w \times$ temperature difference $\times$ wall area

$0.36 \times 16 \times [(12 \times 15) - 50] = 750$ Btu per hr

Windows

$1.1 \times 16 \times 50 = 880$ Btu per hr

Inside Walls

$0.27 \times 13 \times [(2 \times 20 \times 12) + (15 \times 12)] =$ 2,310 Btu per hr

Floor and Ceiling

$0.32 \times 13 \times (2 \times 15 \times 20) = 2{,}500$ Btu per hr

Total conduction due to temperature difference at 3 p.m. $=$ 6,440 Btu per hr.

Heat Gain from Ventilation Air

Ventilation air, brought in to replace the vitiated indoor air and to eliminate odors, must be cooled to room temperature and dehumidified to room humidity. The sensible and latent heat gains from ventilation air are determined accurately from the following two formulas:

Sensible Heat, Btu per hr $=$

$$\frac{Q \times 60}{v} \times 0.240 \times (t_o - t_i)$$

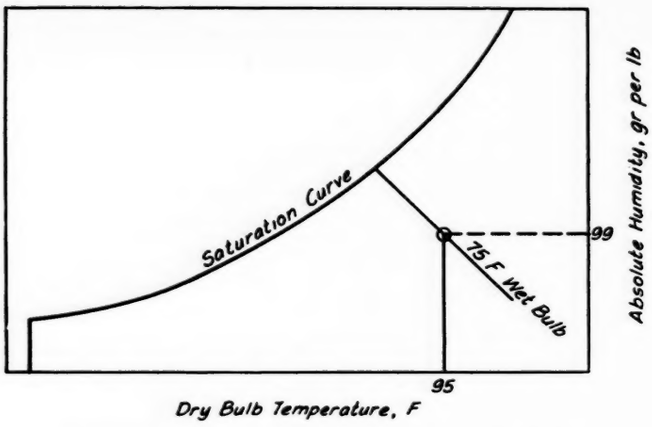


Fig. 1. Method of using a psychrometric chart.

Latent Heat, Btu per hr =

$$\frac{Q \times 60}{v} \times \frac{w_o - w_i}{7{,}000} \times 1{,}060$$

where

- Q = cubic feet per minute of ventilation air
- v = specific volume of ventilation air in cubic feet per pound
- t_o = outdoor dry bulb temperature
- t_i = indoor dry bulb temperature
- w_o = absolute humidity of outdoor air in grains of moisture per pound of dry air
- w_i = absolute humidity of indoor air.

For all practical purposes, these two formulas can be simplified to the following:

Sensible heat, Btu per hr = $Q \times (t_o - t_i) \times 1.08$
Latent heat, Btu per hr = $Q \times (w_o - w_i) \times 0.68$

To determine the moisture gain in pounds per hour divide the latent heat gain, Btu per hr by 1,060.

The various terms involved in these two formulas are easily determined. An explanation has already been given, for example, concerning the determination of the design temperature difference, $t_o - t_i$.

The outdoor absolute humidity, w_o, is found from the design dry bulb and wet bulb temperatures (given in Tables 2 and 3) and from a psychrometric chart. Fig. 1 shows how the psychrometric chart is used in this connection for a dry bulb of 95F and a wet bulb of 75F, which correspond to an absolute humidity, w_o, of 99 gr per lb of dry air.

The indoor absolute humidity, w_i, to be used as a basis for design is given in Table 3 for various classes of comfort cooling applications. In the case of industrial and other special applications, the requirements of the job will ordinarily determine what dry bulb temperatures and relative humidities must be provided. With the aid of the psychrometric chart, the absolute humidity, w_i, may then be determined.

The only term remaining to be explained is Q, the quantity of ventilation air supplied. There are, in general, two ways by which outside air may enter an enclosure:

(1) It may be brought in under control by a fan or blower.

(2) It may be allowed to drift in haphazardly through open windows or window cracks.

Mechanical ventilation—If mechanical ventilation is used, the amount of air brought in should be adjusted to the actual requirements of the occupants. These requirements are somewhat flexible, but available evidence seems to indicate that the figures given in Table 4 represent good practice. They should be used in calculations of this sort.

Natural infiltration—If dependence is placed upon natural infiltration to supply outside air, it is necessary to determine whether the rate of infiltration is great enough to supply the requirements listed in Table 4. If it is not sufficient, then it may be assumed that the occupants will raise a window or open a door to increase the rate of air supply. Table 5 gives approximate data on the rate of natural infiltration for various types of rooms. Hence, for cases where ventilation air is supplied by this means, the following procedure should be used in calculation:

(1) Calculate the rate of infiltration (see Table 5) from the formula

$$\text{(Infiltration, cfm)} = \frac{\text{(volume of room)} \times \text{(air changes per hour)}}{60}$$

(2) Calculate the total amount of outside air required from average number of occupants and Table 4.

TABLE 4—VENTILATION AIR STANDARDS

Application	Cfm Per Person		Per cent of People Assumed Smoking
	Recommended	Minimum	
Apartments	15	10*	Some
Art galleries	1 change per hr	1 change per hr	None
Assembly rooms	10	5	None
Banks	1 change per hr	1 change per hr	Occasional
Barber shops	15	7	Occasional
Beauty parlors	15	7	Occasional
Cafeterias	15	10	Occasional
Cigar stores	15	7	Occasional
Cocktail bars	20	10	30%
Coffee shops	15	7	Occasional
Dance halls	20	10	Occasional
Dept. stores	7	5	None
Drug stores	15	7	Occasional
Funeral parlors	10	5	None
Hospital rooms	15	10	None
Operating rooms —hospitals	100% fresh air	100% fresh air	None
Hotel rooms	15	10*	Some
Offices, general	15	10	Some
Offices, directors rooms	30	20	80%
Offices, private	30	15*	50%
Pool rooms	20	10	50%
Pressing stores	20	10	None
Residences	Use actual infiltration	Use actual infiltration	Some
Restaurants	15	12	25%
Shops, retail	10	7	None
Theatres	7½	5	None

*Figure at least five people for a single medium sized room and eight people for a larger room, but do not pyramid such loads for multiple rooms beyond the maximum simultaneous peak.

Regardless of the CFM per person recommended, the ventilation air quantity should not be less than natural infiltration through windows, doors, and cracks.

These values are given as a guide. The customers' wishes regarding ventilation should always be given serious consideration.

(3) Use the larger of the figures found in (1) and (2).

As in the case of the heat conduction through walls, calculations of heat gain from outdoor air made in the manner outlined give, automatically, values which obtain at 3 p.m. This is because the maximum air temperature during the day occurs at approximately that hour. Inasmuch as there are many cases where the conditions are such that the maximum total heat gain from all sources occurs at some other hour (due to the preponderant effect of a morning sun or heavy occupancy) it may frequently be necessary to make the calculation for some time other than 3 p.m., when the air temperature will be somewhat less than the selected design value. This can be done by making a correction.

It must be noted, however, that the correction applies only to the sensible heat gain, and not to the latent heat gain, because the outdoor absolute humidity ordinarily remains practically constant while the temperature is going through its cyclic variation. This is all handled correctly and simply on the heat gain calculation sheet to be described.

As an example of how this component of the cooling load is calculated, the office described on page 85 is used.

(1) From Table 5 the natural infiltration is equivalent to approximately one air change per hour.

$$\text{Room volume} = 15 \times 20 \times 12 = 3{,}600 \text{ cu ft}$$

$$\text{Infiltration} = \frac{3600 \times 1}{60} = 60 \text{ cfm.}$$

(2) From Table 4 the recommended ventilation air for an office is 15 cfm per person or a total of 45 cfm for three persons. Since the natural infiltration of 60 cfm is greater than the required ventilation of 45 cfm, use 60 cfm in determining the heat gain.

(3) The sensible heat gain at 3 p.m. due to outside air $= 60 \times (96 - 80) \times 1.08 = 1{,}040$ Btu per hr.

(4) Assume an outdoor design wet bulb temperature of 77F. The corresponding absolute humidity, with a 96F dry bulb is 109 gr per lb dry air.

(5) The indoor design conditions are 80F D.B. and 68F W.B., which correspond to an absolute humidity of 84 gr per lb of dry air.

(6) The latent heat gain due to outside air $= 60 \times (109 - 84) \times 0.68 = 1{,}020$ Btu per hr.

(7) The total heat gain, at 3 p.m., due to outside air $= 1{,}040 + 1{,}020 = 2{,}060$ Btu per hr.

Calculation of Heat from Occupants

Heat given off by persons varies with the degree of activity of the individual. Precise data on this point have been collected over the years by many authorities. Typical examples are occupants of theaters, 390 Btu per hr; clerks in stores, 550 Btu per hr; factory workers, 1,000-1,500 Btu per hr.

In most air conditioning applications people are found in one or two general states of activity—either they are seated or standing at rest, moving about only occasionally, as in a home or an office, or they are engaged in brisk activity such as walking or dancing

TABLE 5—NATURAL INFILTRATION RATES FOR VARIOUS TYPES OF ROOMS

Kind of Room	Number of Complete Air Changes per Hour
Rooms—1 side exposed	1
2 sides exposed	1½
3 sides exposed	2
4 sides exposed	2
No windows or outside doors	½ to ¾
Entrance halls	2 to 3
Reception halls	2
Bathrooms	2
Drug stores	2 to 3
Clothing stores	1
Churches, factories, lofts, etc.	½ to 3

(for example, restaurant waiters or night club patrons). The heat gain and moisture gain from people under such circumstances is as follows:

(1) Average condition (at rest, eating, office work, etc.)

Sensible heat — 220 Btu per hr.
Latent heat — 180 Btu per hr.

Total — 400 Btu per hr.

$$\text{Moisture gain} = \frac{180}{1{,}060} = 0.17 \text{ lb per hr}$$

(2) Medium rate of exertion (restaurant waiters, average dancing, etc.)

Sensible heat — 230 Btu per hr
Latent heat — 420 Btu per hr

Total — 650 Btu per hr

$$\text{Moisture gain} = \frac{420}{1{,}060} = 0.40 \text{ lb per hr}$$

Consideration should, of course, be given in all cases to the number of people who fall in one category or the other.

In many applications such as offices, the people involved are present throughout the day and there is no problem in selecting the time of day for making the calculation. In other cases, such as restaurants, the occupancy changes sharply at certain hours of the day and a question arises as to the time of day to be used in the calculation. This matter is discussed later under the heading, Selecting the Time of Day.

In the office design example, the heat gain from occupants of the office is as follows:

Sensible heat $= 3 \times 220 =$ 660 Btu per hr
Latent heat $= 3 \times 180 =$ 540 Btu per hr

Total heat = 1,200 Btu per hr

Heat from Lights and Appliances

From the standpoint of the heat gain calculation, heat dissipating appliances which give off only sensible heat may be divided into three classes:

(1) Appliances with known electrical watts input, or where it can be read from a nameplate.
(2) Appliances with electrical watts input that can be determined through brief calculation.
(3) Appppliances which have no electrical input.

In the first category may be found such devices as lamps, toasters, waffle irons, etc., where the nameplate indicates the electrical current consumption to be expected. All this current, which is changed into heat, is given off to the air in the room. The amount of heat, Btu per hr thus liberated is equal to 3.4 times the watts input. One watt is equivalent to 3.415 Btu per hr.

Under the second heading may be included devices like motors, which show on their nameplate a horsepower rating but do not give the watts input for that rating. The watts input depends on the motor efficiency, which may range from 50 per cent for small, single phase, fractional horsepower motors, up to about 85 per cent for moderate sized and large three-phase induction motors. The rated watts input may be calculated from the formula—

$$\text{Watts input} = \frac{\text{motor horsepower} \times 746}{\text{efficiency (as a decimal)}}$$

If the load which the motor is driving is wholly within the room being air conditioned, then the entire watt input of the motor must be figured into the heat gain, using the conversion factor 3.4 to convert watts into Btu per hr. If, however, the motor drives a load located in another room (through a long shaft or belt, for example) then the energy transmitted to the load must not be included in the heat gain. In such cases the heat given off to the room consists only of the motor losses, and may be calculated from the formula—

Heat from motor, in Btu per hr = (1 — motor efficiency expressed as a decimal) × (watts input to motor) × 3.4

If there is no way of determining accurately the motor efficiency, then as a rough approximation the heat gain from such sources can be obtained by using the data in Table 6, multiplying the figures in the table by the motor horsepower rating.

The third class of appliances takes in items like ranges, steam tables, coffee urns, uncovered steam pipes, etc., where no electrical power consumption is involved. For such cases, while a number of heat gain tables are available, it is best to write the manufacturer mentioned on the nameplate for the rating of the equipment being considered.

Whenever dealing with heat generating appliances consideration should be given to special hoods and exhaust fans which can be arranged to exhaust some air from the spaces around the appliances and carry away a part of the heat directly to outdoors, thus reducing the load on the air conditioning system. Some of the heat from appliances is, of course, in the form of radiation or heat rays and can not be removed by the exhaust air unless it is trapped within the hooded enclosure by screening partitions. This radiation strikes walls and objects in the air conditioned space and is finally given off to the air, hence it must be included in the heat gain.

In many other cases it will be possible to estimate with reasonable accuracy the percentage of the heat generated by the device that is carried away by the ventilating system and does not get into the room to increase the cooling load. If such an estimate cannot be made, use the following procedure to determine heat gain from hooded devices:

(1) Obtain the air delivered by the fan through the hood in cfm.
(2) Obtain the temperature of the air carried up the hood, and also the room temperature.
(3) Calculate the heat carried up the hood, from the following formula:

Heat carried up hood, in Btu per hr =
(Cfm of air through hood) × (T_h — T_r)

where

T_h = temperature of air carried up the hood.
T_r = room temperature.

(4) Subtract the heat carried up the hood from the total heat generated by the device. The difference is the heat gain which is added to the room.

In applying hoods to heat generating devices, care must be taken not to increase the heat gain (through increased outside air supply) more than it is reduced. If more air must be exhausted through the hood than is normally required for ventilation, it may be desirable to bring a supply of unconditioned air from outdoors, through a special duct, and discharge it into the hooded area. This eliminates the drain of cooled air from the conditioned space.

There are some appliances which give off not only sensible heat, but latent heat as well. These will gen-

TABLE 6—HEAT GENERATED BY MOTORS AND MOTOR GENERATORS OPERATING CONTINUOUSLY

Figures are based on nameplate rating. Use where actual watts input cannot be determined.

Nameplate Rating in HP	Motors			Motor Generators	
	Connected Load in Air Conditioned Space, Motor Outside	Motor in Air Conditioned Space, Load Outside	Motor and Connected Load Both in Air Conditioned Space	Nameplate Rating in KW	Heat Gain in Btu per Hr per KW Rating
1/20 to ⅛	2546	2354	4900	¾ to 3	2800
⅛ to ½	2546	1374	3920	3 to 5	1300
½ to 3	2546	724	3270		
3 to 20	2546	454	3000		
20 to 150	2546	284	2830		

erally be encountered in industrial applications. Any device which evaporates moisture into the air falls into this category, and the moisture thus evaporated must be included in the calculations. The calculation is as follows:

Latent heat, Btu per hr =
(moisture, lb per hr) × 1060

It is clear from this formula that one must first ascertain the amount of moisture being evaporated.

A pitfall to avoid in this calculation is the inclusion of latent heat from appliances into two different parts of the calculation. Thus, suppose an electric hot plate rated at 500 watts were used to boil water, resulting in the evaporation of a half pound an hour. The total heat dissipated by the hot plate would be 500 × 3.4 = 1,700 Btu per hr. The latent heat given to the air would be ½ × 1060 = 530 Btu per hr. The thing to note is that the 530 Btu is included in the 1,700 Btu given off by the hot plate. Thus

Total heat from hot plate = 1700 Btu per hr
Latent heat from hot plate = 530 Btu per hr

Sensible heat from hot plate = 1170 Btu per hr

In the design example of the office, the only appliance load is the two electric lights which are used during the daytime. Hence this component of heat gain is

200 × 3.4 = 680 Btu per hr.

Sun Effect Through Walls

As has already been mentioned, the sun transmits invisible heat rays which fall on exposed surfaces and force heat through walls and windows into occupied structures. This heat gain is in addition to the heat leakage which is produced by the temperature difference between outside and inside air, and must be calculated separately. The method used in making calculations for wall structures is somewhat different from windows; both cases will be discussed.

The amount of solar heat which strikes a wall can be simply and accurately measured by means of an instrument known as a pyrheliometer. Many measurements have been made all over the world and considerable data on this subject is available. However, not all the heat which strikes a wall penetrates through it and the problem faced in the heat gain calculation is to determine what per cent of the heat deposited on the outside surface actually gets through to the air inside the room.

It must be noted that part of the heat which strikes the wall is reflected back into space, Fig. 2, because all surfaces act as heat mirrors to a greater or lesser extent. Highly polished metallic surfaces, for example, are very good heat mirrors and may reflect as much as 90 per cent of the heat. Rough black surfaces, on the other hand, are rather poor heat mirrors and may reflect only 10 per cent of the heat. In any event the nature of the outside surface of the wall is certainly an important factor in determining how much solar heat will penetrate through the wall.

That part of the solar heat which is not reflected is absorbed on the outer surface of the wall. Only a fraction of the heat which is thus absorbed, however, succeeds in forcing its way through the wall and into the conditioned room, because the absorption of heat on the outer surface tends to increase the surface temperature, and this rise in surface temperature results in a loss of heat to the outside air by conduction, convection and radiation. The amount of heat which is finally transmitted through the wall and into the room depends on the heat transmission characteristics of the wall; that is to say, on the heat transmission coefficient U_w.

To recapitulate, it may be said that the additional heat gain from the sun through exposed walls depends on three factors:

(1) The intensity of solar radiation on the outside surface of the wall.
(2) The absorption coefficient of the outside surface of the wall.
(3) The overall heat transmission coefficient U_w.

In making heat gain calculations, these three factors are combined in a single radiation coefficient R_w, which is found from the following formula:

$$R_w = I \times a \times F$$

where

I = intensity of solar radiation, or the amount of heat in Btu per hr from the sun, striking each square foot of the wall.
a = absorption coefficient of the outside surface of the wall.
F = percentage (expressed as a decimal) of the sunshine energy absorbed by the wall which is conducted to the inside.

The radiation coefficient R_w, thus calculated, gives the additional heat in Btu per hr which passes through every square foot of exposed wall surface, due to the action of the sun. It remains to be explained how values of the three factors — *I*, *a* and *F* may be found for insertion into the above formula.

The intensity of solar radiation *I*, is not a fixed quantity. It depends on the latitude of the city for which calculations are being made, on the time of day, and on the direction in which the wall faces. Table 7 gives a complete summary of values of this factor, which are to be used in heat gain calculations. Values are given for different latitudes; use the section of the table nearest the latitude of the application. These values were calculated from sun's intensity tables and contain corrections for window size, building over-

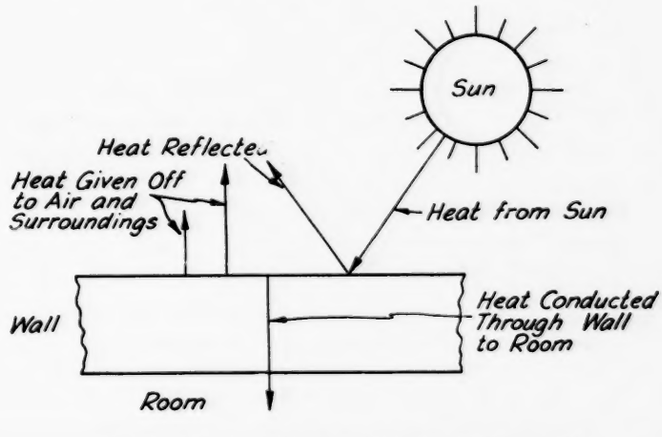


Fig. 2. Action of sun's heat on building wall.

TABLE 7—HEAT IMPINGING ON WALLS, ROOFS AND GLASS FOR VARIOUS NORTH LATITUDES

(Data apply to Aug. 1. Figures in Time of Day column are a.m. from 6 to 11, p.m. from 1 to 6)

WALLS AND ROOFS

Degrees North Lat	Time of Day	Direction Wall Faces NE	E	SE	S	SW	W	NW	Horizontal Surface
		Value of I in Btu per hr Striking 1 sq. ft.							
30	6	37	47	23	5	5	5	5	11
	7	119	145	91	11	11	11	11	64
	8	153	207	149	17	17	17	17	147
	9	130	194	158	35	21	21	21	213
	10	86	152	143	63	24	24	24	262
	11	35	94	85	80	26	24	26	290
	Noon	26	26	65	85	65	26	26	300
	1	26	26	26	80	85	94	35	290
	2	24	24	24	63	143	152	86	262
	3	21	21	21	35	158	194	130	213
	4	17	17	17	17	149	207	153	147
	5	11	11	11	11	91	145	119	64
	6	5	5	5	5	23	47	37	11
35	6	43	49	27	5	5	5	5	13
	7	121	151	97	11	11	11	11	72
	8	147	207	155	25	17	17	17	151
	9	120	194	169	49	21	21	21	213
	10	71	152	156	83	24	24	24	245
	11	28	94	129	103	26	26	26	288
	Noon	26	26	84	109	84	26	26	298
	1	26	26	26	103	129	94	28	288
	2	24	24	24	83	156	152	71	245
	3	21	21	21	49	169	194	120	213
	4	17	17	17	25	155	207	147	151
	5	11	11	11	11	97	151	121	72
	6	5	5	5	5	27	49	43	13
40	6	49	56	32	5	5	5	5	20
	7	123	162	109	11	11	11	11	85
	8	137	211	166	29	17	17	17	160
	9	102	195	181	74	21	21	21	212
	10	54	152	171	103	24	24	24	244
	11	28	94	144	124	41	26	26	281
	Noon	26	26	98	128	98	26	26	290
	1	26	26	41	124	144	94	28	281
	2	24	24	24	103	171	152	54	244
	3	21	21	21	74	181	195	102	212
	4	17	17	17	29	166	211	137	160
	5	11	11	11	11	109	162	123	85
	6	5	5	5	5	32	56	49	20
45	6	87	99	56	6	6	6	6	27
	7	151	192	134	12	12	12	12	89
	8	144	237	188	48	17	17	17	156
	9	100	199	197	93	21	21	21	205
	10	46	153	184	121	24	24	24	243
	11	28	94	158	146	63	26	26	259
	Noon	26	26	116	156	116	26	26	281
	1	26	26	63	146	158	94	28	259
	2	24	24	24	121	184	153	46	243
	3	21	21	21	93	197	199	100	205
	4	17	17	17	48	188	237	144	156
	5	12	12	12	12	134	192	151	89
	6	6	6	6	6	56	99	87	27

WINDOWS AND SKYLIGHTS

Degrees	Time	Value of R_g in Btu per hr Striking 1 sq. ft. of Single Glass. Multiply by 0.9 for Double Glass.							
30	6	31	40	15	3	3	3	3	7
	7	95	123	63	8	8	8	8	47
	8	112	174	104	12	12	12	12	122
	9	81	154	108	16	15	15	15	190
	10	44	101	92	29	16	16	16	236
	11	19	51	50	41	18	18	18	263
	Noon	18	18	32	43	32	18	18	272
	1	18	18	18	41	50	51	19	263
	2	16	16	16	29	92	101	44	236
	3	15	15	15	16	108	154	81	190
	4	12	12	12	12	104	174	112	122
	5	8	8	8	8	63	123	95	47
	6	3	3	3	3	15	40	31	7
35	6	36	42	18	3	3	3	3	8
	7	96	129	68	8	8	8	8	54
	8	104	174	112	12	12	12	13	126
	9	71	153	121	22	15	15	15	190
	10	34	101	105	42	16	16	16	220
	11	18	51	79	57	18	18	18	261
	Noon	18	18	43	62	43	18	18	270
	1	18	18	18	57	79	51	18	261
	2	16	16	16	42	105	101	34	220
	3	15	15	15	22	121	153	71	190
	4	13	12	12	12	112	174	104	126
	5	8	8	8	8	68	129	96	54
	6	3	3	3	3	18	42	36	8
40	6	41	48	21	3	3	3	3	13
	7	96	138	78	8	8	8	8	64
	8	93	176	124	12	12	12	12	133
	9	58	153	135	35	15	15	15	188
	10	25	112	121	56	16	16	16	219
	11	18	50	93	74	21	18	18	254
	Noon	18	18	53	79	53	18	18	263
	1	18	18	21	74	93	50	18	254
	2	16	16	16	56	121	112	25	219
	3	15	15	15	35	135	153	58	188
	4	12	12	12	12	124	176	93	133
	5	8	8	8	8	78	138	96	64
	6	3	3	3	3	21	48	41	13
45	6	72	85	37	3	3	3	3	18
	7	116	165	98	8	8	8	8	67
	8	94	198	144	18	12	12	12	129
	9	58	155	151	47	15	15	15	181
	10	19	100	137	71	16	16	16	219
	11	18	50	106	93	30	18	18	234
	Noon	18	18	66	104	66	18	18	254
	1	18	18	30	93	106	50	18	234
	2	16	16	16	71	137	100	19	219
	3	15	15	15	47	151	155	58	181
	4	12	12	12	18	144	198	94	129
	5	8	8	8	8	98	165	116	67
	6	3	3	3	3	37	85	72	18

hang, and absorption and reflection by the glass. The manner of selecting the proper time of day to be used in this calculation will be discussed.

Values of the absorption coefficient, a, for various types of surfaces, are given in Table 8. It will be noted that very light colored surfaces have an absorption coefficient of 0.4, which means that they absorb only 40 per cent of the solar heat which strikes them and reflect 60 per cent of this heat. Very dark colored surfaces have an absorption coefficient of 0.9, which indicates an absorption of 90 per cent of the solar heat and a reflection of only 10 per cent. Values in this table are the result of numerous careful experiments that have been conducted in scientific laboratories all over the world and may be used safely in making calculations of this sort.

Values of the radiation factor F can be secured from the formula

$$F = 0.167\ U_w$$

F depends entirely on the wall transmission coefficient U_w, which is always determined early in the calculation when figuring the heat leakage through the wall due to temperature differences.

It should hardly be necessary to point out here that these calculations of heat gain from the sun are made only when the wall for which calculations are being made is definitely exposed to the sun. If the wall is shaded by trees or other buildings, then there will be no sunshine energy striking the wall directly and the calculation of this factor should be omitted. When the wall is only partly shaded, it will be necessary to estimate what percentage of the wall is thus protected and to make allowances accordingly.

Certain special cases arise when the rooms for which calculations are being made are located under attics or under flat or sloping roofs. Table 1 gives values of R_w to use in such cases.

For an example, the additional conduction due to sun effect on the south wall at 1 p.m. is figured for the office described.

$F = 0.06$ (for $U_w = 0.36$)
$a = 0.7$ (from Table 8 for a brick wall)
$I = 124$ (from 40 deg N. latitude — section of Table 7)
$R_w = 0.06 \times 0.7 \times 124 = 5.20$

(Conduction due to sun effect) $= 5.20 \times (12 \times 15) - 50 = 6.75$ Btu per hr. Note that the conduction due to temperature difference alone was only 750 Btu per hr for this same wall.

Sun Effect Through Windows

Transparent windows present a problem somewhat different from that of opaque walls, because they permit about 90 per cent of the energy of the sun to pass through the window just as though no obstacle existed. The exact percentage depends on the angle of the sun's rays, the thickness of the glass, etc. In making calculations for such windows, therefore, it is not necessary to consider any surface absorption coefficients or any heat transmission coefficients. Having a knowledge of the direction in which the window

faces at the time of day for which calculations are being made, it is possible to ascertain directly the amount of additional heat in Btu per hr which the sun passes through each square foot of window area. Table 7 summarizes the values of this solar radiation coefficient, designated as R_g, to be used for single and double glass windows, having various orientations, in four different latitudes and at various times of day. The total solar radiation through a window, in Btu per hr, is determined by multiplying R_g by the area of the window, in square feet. The proper time of day to use in calculations will be discussed later. In the case of skylights, the values given in Table 7 for horizontal surfaces should be used.

When windows are completely shaded from the sun by trees or other buildings, then none of the solar energy passes through them. In the absence of such accidental shading, there are various man-made devices such as awnings, shades, and Venetian blinds which may be used to mitigate the effect of the sun to a greater or lesser extent.

The reduction in sun effect heat gain through windows equipped with various types of awnings and shades is as follows:

With a dark outside shading screen, the heat gain from the sun is reduced 65 to 80 per cent; canvas awnings, 65 to 75 per cent; light colored outside Venetian blind, 50 to 65 per cent; aluminum finish inside Venetian blind, 20 to 35 per cent; light color roller shade, half drawn, 30 per cent; dark roller shade, half drawn, 5 to 10 per cent.

For example, the direct sun effect, at 1 p.m., through the windows of the office in the problem, is figured.

Direct sun effect $= 74 \times 50 = 3{,}700$ Btu per hr. where

50 = sq ft of window area exposed to sun at 1 p.m.

74 = Radiation coefficient, R_g, for 40 deg N. latitude at 1 p.m.

If dark awnings are installed to protect this window from the sun, the heat gain from the sun is reduced 65 to 75 per cent. Then the actual sun effect $=$ $3700 - (3700 \times 0.70) = 1{,}110$ Btu per hr.

Combining the Heat Gain Components

Up to this point the discussion has centered on methods of calculating each individual heat gain component, and it has been stated that attention must be directed to the time of day for which calculations are made. The reason for this is apparent when it is considered that all the factors which contribute to the heat gain vary widely in magnitude during the course of a day.

Consider, for example, the outside air temperature, which is the causative factor in the elements of heat conduction through walls and sensible heat gain from outside air. During a typical 24-hour period in summer, the outside air temperature might go through a cyclical variation with the low point at 5 a.m., and the high at 3 p.m. Although a typical daily outdoor maximum variation may be only 15 deg between 5 a.m. and 3 p.m., in some localities, like Arizona, for example, this variation might frequently be of the order of 40F or more. It will be clear from this figure that the magnitude of the heat gain components which depend on outside temperature will be different at various hours of the day.

TABLE 8—SOLAR ABSORPTION COEFFICIENTS FOR VARIOUS BUILDING MATERIALS

Type Surface	Absorption Coefficient, a
Very light colored surfaces as—	
White stone	
Very light colored cement	0.4
White or light cream colored paint	
Medium dark surfaces as—	
Asbestos shingles	
Unpainted wood	
Brown stone	0.7
Brick and red tile	
Dark colored cement	
Stucco	
Red, green or gray paint	
Very dark colored surfaces as—	
Slate roofing	
Tar roofing materials	0.9
Very dark paints	

The latent heat gain from the outdoor air is not appreciably affected by these temperature fluctuations, since the absolute humidity outdoors remains more or less constant during any given day. That is, the relative humidity goes through a cyclical variation which is just the opposite of the temperature cycle. When the temperature is highest, the relative humidity is lowest, and vice versa.

The solar radiation intensity also goes through some interesting variations, and involves not only the time of day but also the orientation of exposed surfaces. Reference to Table 7 will show, for example, that on walls facing in various directions, the sun reaches its maximum intensity at the following hours: East wall, 8 a.m.; south wall, noon; west wall, 4 p.m.; horizontal surface, noon. On east walls, the sun strikes only in the morning. On west walls, the sun strikes only in the afternoon. The angle of the sun's rays is continually changing.

Knowing the specific character of the variation in temperature, solar radiation, occupancy, and other factors, it is possible on any particular job to draw curves showing the variation in all the components which enter into the heat gain. Fig. 3 shows such a family of curves, worked up in the calculation of heat gain for a residence, which takes into account the fact that the outdoor temperature and the solar intensity have different values at different times of the day. In this figure the components have been added in their proper phase relation to each other to give the total heat gain, which is shown in the last of the five curves.

An analysis of Fig. 3 shows that whereas individual components attain peak values at 7 a.m., 11 a.m., 1 p.m., 3 p.m., and 5 p.m., respectively, the total heat gain is a maximum at 4 p.m. This means that if one were making a calculation of heat gain for this structure and did not wish to go through the lengthy procedure that was followed in drawing up these curves, the cal-

culation of all components should be made for 4 p.m. The value of maximum heat gain thus determined would agree with that shown in Fig. 3.

Clearly it would not be accurate, though it would be conservative, to find the greatest value that each component attained during the day, and then add all these greatest values together to find the maximum total heat gain. The method of calculation used should be sufficiently flexible to enable taking advantage of the fact that when one component is at its peak, other components have either passed or have not yet reached their peaks. The introduction of this flexibility and accuracy into the calculations requires that they be made for a predetermined time of day in each case, and that this time of day be selected with care.

How this may be done is answered in the following paragraphs.

Selecting the Proper Time of Day

In selecting the time of day for calculations one must distinguish two general types of application. The first class includes all applications which do not have an unusually large and sharply variable heat gain from people or appliances. In this first class would be included applications in residences, offices, and some types of retail shops. The second class would include all applications which do have a large heat gain from people and appliances, varying sharply at certain hours of the day. In this category would fall such applications as restaurants, funeral parlors, night clubs, etc.

Cases in the first category may be handled quite simply and with reasonable accuracy. The time of maximum heat gain in such cases is dependent mainly upon climatic factors and degree of exposure of the space being air conditioned, and not so much on variables like occupancy and heat producing appliances. Because of this it has been possible to make a study of the probable time of maximum heat gain in such spaces with various exposures, and to reach substantially accurate conclusions beforehand. The result of this study is given in Table 9. With the aid of this data it is possible to determine immediately the approximate time of day to be used in the calculations.

Table 9 was worked up on the basis that the rooms under consideration were approximately square; that outside walls were exposed to the sun, and were not shaded by other buildings; that outside walls had one or more windows; that the heat gain from occupants and appliances was small compared to the heat gain from other sources. Consequently this data does not apply exactly to rooms which do not conform to these conditions, although it is quite correct for rooms which do not deviate too widely from these specifications.

It is not very difficult to select the proper time of day for any room. For example, consider a room which has a south and west exposure. Take the case where there are no awnings on the windows, and an occupied space above, such as might occur in a modern office building. Sun effect on south wall and windows reaches a maximum at noon (Table 7). The sun effect on the west wall and windows reaches a maximum at 4 p.m., and the value thereof is greater than for the south wall. The outdoor temperature, and consequently the tendency to gain heat by conduction through practically all walls, ceiling, floor, etc., reaches a maximum about 3 p.m. The heat gain from outdoor air also reaches a maximum about 3 p.m. From this it is apparent that the effects at 3 p.m., or thereabouts, will overshadow the effects at noon. Furthermore, the sun effect on the west wall at 3 p.m. will be almost as great as at 4 p.m. Consequently 3 p.m. is the proper time for which to make the calculations.

Now suppose that this same room were located under a roof or a hot attic. The sun effect above the room would be a large factor and would reach a maximum at noon, just the same as on the south wall. The effect of this would be to make the maximum total heat gain come somewhat earlier than before—probably at 2 p.m. instead of 3 p.m., as shown in Table 9.

Suppose now that this room were a restaurant doing its greatest business at dinner, and therefore having

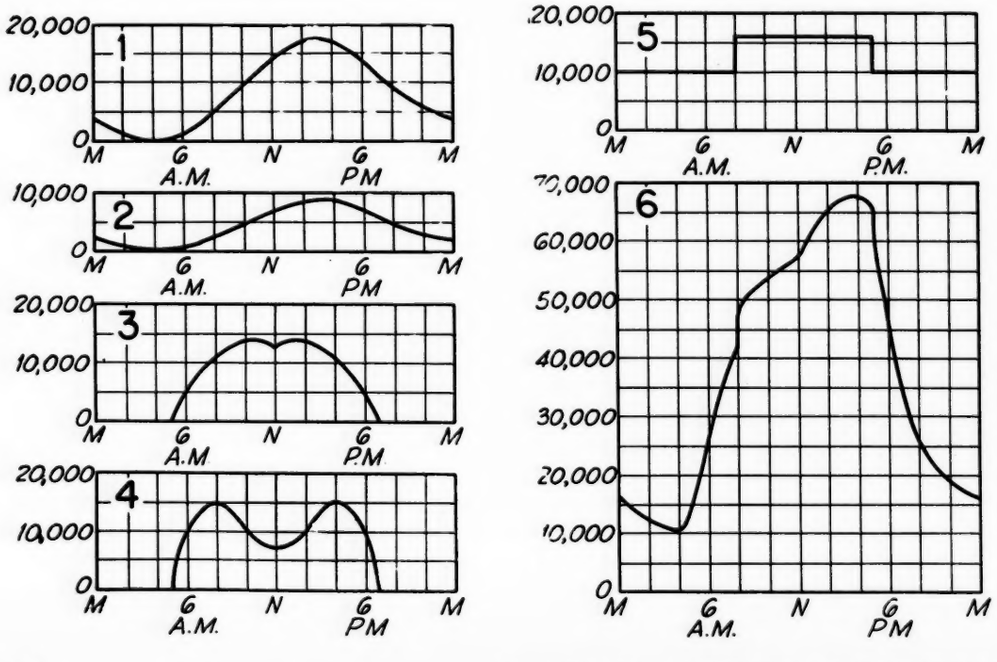


Fig. 3. Time variations of the components and total of cooling load in a suite of offices. (1) Conduction caused by outdoor-indoor temperature difference. (2) Sensible heat from ventilation air. (3) Additional conduction through east, south and west walls and roof, caused by heat received from the sun. (4) Load imposed by direct sunshine through windows. (5) Heat from people, and latent heat from ventilation air. (6) Total of preceding components.

its maximum occupancy between the hours of 6 and 8 p.m. The people present would naturally supply a considerable proportion of the total heat gain. It would not be correct to make the calculations for 2 or 3 p.m. and then add the heat gain from occupants, which occurs at the later hour. On the other hand, it would not be quite accurate to make calculations of all components for 6 p.m., because of the fact that conduction and sun effect through walls experience a time lag or retardation and manifest their maximum effect indoors several hours after the causative elements (outdoor air temperature, and sun intensity) have attained their maximum values. The correct solution in this case would be to calculate the values of all components, excepting occupancy, about 4 or 5 p.m., and add to them the maximum heat gain from occupants.

On the other hand, if the restaurant were one doing its greatest business at the lunch hour, the maximum heat gain would most likely occur between noon and 1 p.m.

Needless to say, all cases which cannot be handled by the standard method embodied in Table 9 must be given careful consideration. In case of serious doubt as to the proper time to use, one might proceed as follows:

(1) Calculate the maximum value which each component attains during the day, assuming there is no time lag in the heat transmission through walls.
(2) Note the hours at which these maximum values occur.
(3) Give consideration to time lag phenomena in walls, in a manner to be discussed later.
(4) Select the probable time of maximum total heat gain from an analysis of the above data.

Example: The office described on page 85 has one wall exposed on the south. From Table 9, the proper time of day for calculation is 1 p.m. It would, therefore, be necessary to correct the heat gain from conduction through walls and the sensible heat gain from outdoor air, to this time of day. The heat gain from sun effect has already been calculated for 1 p.m.

Note that in all this discussion the time of day is local sun time. Tables 7, 9 and 10 are based on local sun time. For practical purposes, however, these tables may be used for local standard time.

TABLE 9—TIME OF DAY WHEN MAXIMUM COOLING LOAD OCCURS IN ROOMS WITH DIFFERENT TYPES OF EXPOSURE

Occupancy and appliances are neglected

Number of Walls Exposed	Direction of Exposed Wall or Walls	Awnings on Windows		No Awnings on Windows	
		Occupied Space Above	Roof or Attic Above	Occupied Space Above	Roof or Attic Above
1	N	--	--	2	2
	NE	--	--	2	2
	E	2	2	9	9
	SE	1	1	10	10
	S	2	1	1	1
	SW	3	2	4	3
	W	3	3	4	4
	NW	4	3	5	4
2	N E	2	2	9	9
	NE SE	2	1	9	9
	E S	2	1	10	10
	SE SW	3	2	3	3
	S W	3	2	3	3
	SW NW	3	3	4	4
	W N	3	3	4	4
	NW NE	4	3	5	5
3	W N E	4	3	4	4
	NW NE SE	3	3	4	4
	N E S	2	2	10	10
	NE SE SW	3	2	3	3
	E S W	3	2	3	3
	SE SW NW	3	3	4	4
	S W N	3	3	4	4
	SW NW NE	4	3	4	4
4	S W N E	3	2	3	3
	SW NW NE SE	3	2	4	4
None				2	2

All times of day are p.m., except 9 and 10 which are a.m.

Outdoor Temperature at Peak Load

It has already been mentioned that the method described earlier for calculating the heat gain from conduction through walls and from outside air automatically assumes that the time for which calculations are made is 3 p.m., local sun time (or for practical purposes local standard time), if the outdoor temperature is the design maximum outdoor temperature given in Table 2. It has also been pointed out that in many cases it is necessary to correct the outdoor temperature to the time of day at which the peak load occurs, and then to use this outdoor temperature to determine the heat gain at the specified time of day.

The outdoor temperature at the time of peak load depends on the design maximum outdoor temperature (Table 2) and the mean daily temperature range (mean difference between the daily maximum and minimum outdoor temperatures).

The following procedure should be followed:

(1) From design dry bulb and wet bulb temperatures given in Table 2 determine the design outdoor absolute humidity. Use this absolute humidity on the Heat Gain Sheet, together with the design indoor absolute humidity, to calculate the latent heat gain of ventilation air.

(2) From the nearest station of the U. S. Weather Bureau determine the mean daily temperature range for July for that locality.

(3) Determine the outdoor temperature at time of peak load, together with the design indoor temperature to calculate the heat gain from conduction and ventilation air.

The outdoor temperature at the time of peak load is:

$$t_o = t_d - (D \times V)$$

where

t_o = design outdoor temperature at a particular time of day.
t_d = design maximum outdoor temperature (derived from Table 2).
D = an hourly factor, given in Table 10.
V = mean daily temperature range for July (obtained from nearest U. S. Weather Bureau).

The form of Table 10 is planned that when the

TABLE 10.—DESIGN OUTDOOR DRY BULB TEMPERATURES

Design maximum outdoor temperature: $t_d = 96F$
Mean daily temperature range for July: $V = 22F$

Time of Day	Factor D	D × V	Design Outdoor Temperature $t_o = t_d$—(D × V)
1 a.m.	0.8	17.6	78
2	0.9	19.8	76
3	0.95	20.9	75
4	1.0	22.0	74
5	1.0	22.0	74
6	1.0	22.0	74
7	0.9	19.8	76
8	0.8	17.6	78
9	0.65	14.3	82
10	0.5	11.0	85
11	0.3	6.6	89
Noon	0.2	4.4	92
1 p.m.	0.1	2.2	94
2	0	0	96
3	0	0	96
4	0	0	96
5	0.05	1.1	95
6	0.1	2.2	94
7	0.2	4.4	92
8	0.3	6.6	89
9	0.4	8.8	87
10	0.5	11.0	85
11	0.6	13.2	83
Midnight	0.7	15.4	81

proper data are entered, a permanent record is provided of the design outdoor temperature for a particular locality over a 24-hour period. To show how these data are prepared and used, Table 10 was completed using data which apply to a city having an outdoor design dry bulb temperature of 96F, and a mean daily temperature range of 22F for the month of July. Note: In Table 10 fill in the nearest integral number in the column of design outdoor temperatures.

As an example of the method of calculating heat gain from conduction and ventilation air at the correct time of day, note that the time of peak load for the office illustrated has been shown to be 1 p.m. From Table 10 it will be noted that the corresponding design outdoor temperature is 94F.

With a design indoor temperature of 80F this corresponds to a temperature difference of 14F instead of 16F. Hence corrected heat gains, for 1 p.m. are:

Heat from conduction: $14/16 \times 6440 = 5640$ Btu per hr.

Sensible heat from ventilation air: $14/16 \times 1040 = 910$ Btu per hr.

Effect of Heat Storage in Walls

Very little has been said thus far about the subject of heat storage in walls, yet it would seem that it must certainly have some effect on the heat gain of a structure. What is the justification for ignoring it in the method of calculation being described here?

The justification lies in the fact that the method of calculation described on these pages has been compared with some very accurate tests conducted a few years ago on walls having various heat capacities, and has been shown to give highly accurate results for walls which have heat capacities comparable to those found in ordinary building construction. For walls having abnormally high heat capacities, the calculations give results which are slightly on the conservative side. The tests showed that the heat capacity of normal walls has relatively little effect on the amount of heat transmitted through them, but does give rise to a time lag of perhaps 1 to 5 hours, depending on the construction of the wall.

Furthermore, a number of analyses have been made to determine how much difference there would be in the final results if allowance were made for this time lag which is known to exist. The analyses showed in practically every case that there is little difference in the final figures whether the time lag is taken into account or whether it is assumed that no heat capacity, and hence no time lag exists.

It is for these reasons that the heat gain calculation methods here described do not normally contain any heat capacity factors for walls. They are not needed for the usual type of application where the system operates under thermostatic control to maintain the desired temperature.

Let it not be supposed, however, that this matter can be completely ignored. There are a number of cases where one must reckon with this factor if successful installations are to be made. Consider, for example, the problem of air conditioning at night, the bedrooms of a massively built home. Suppose that it is planned to shut down the system in the daytime and to operate it only at night. One might reason: The sun does not shine at night, hence no sun effect need be taken into account; the air temperature outdoors is reduced at night, hence one might assume a smaller temperature difference than in the daytime.

Such reasoning would be fallacious. During the heat of the day, the structure soaks up a considerable amount of heat and the temperature of the walls rises. When the air conditioning system is started in the evening, this heat flows into the rooms and tends to nullify the cooling effect which is produced. In some cases the room air temperature may undergo practically no change for hours after the equipment begins operating. It is necessary to cool the walls before it is possible to cool the air. How can one determine how much cooling capacity is needed in a case like this?

It may be said that if one designs the system so it can carry the maximum daytime load when it is running steadily under thermostatic control, then it should be suitable for the job. If it is operated only at night, then it has some margin of reserve capacity for pull-down. If it is operated all day, then it will certainly keep the place cool at night. This procedure requires, obviously, that the sun effect and high outdoor temperatures during the day be given the usual consideration.

Knowledge on the subject of heat storage in walls and roofs is quite limited. Its effect on the time lag of heat passing through walls and roofs is further limited.

At this stage of the art, heat storage and the time

lag it causes should be neglected for all heat gain calculations except in those cases where the sun effect heat gain through walls and the roof exceed 30 per cent of the total heat gain. Even in this case, time lag should be considered only for sun effect and only for those constructions having a time lag of more than four hours. See Table 11, which is adapted from the ASHVE *Guide*. The effect of time lag on conduction heat gain due to indoor-outdoor temperature difference may be neglected since the temperature differential is maintained for long enough period and the cycle changes slowly enough for its effect to be felt indoors.

If time lag is considered for any wall or the roof, it must be considered for all walls and the roof. It would be unwise and would give a false heat gain to accept the benefit of time lag on a west wall at 3 p.m. and neglect the adverse effect of the late morning sun on a heavy east wall.

In those few instances that time lag is considered, its effect may be calculated by omission of the regular sun effect heat gain calculation for walls and the roof and substitution of the heat gain values obtained by the following procedure.

(1) Determine from Table 11 the time lag in hours for one wall.
(2) Knowing the wall time lag in hours, determine from Fig. 4 the time lag factor, L_t, for each of several hourly periods, previous to the time of day for which the heat gain is being calculated.
(3) Determine the sun effect coefficient, I, for the same hourly periods used in (2).
(4) Multiply the sun effect coefficient, I, by the time lag factor, L_t, and select the maximum as the one to use in the heat gain calculation.

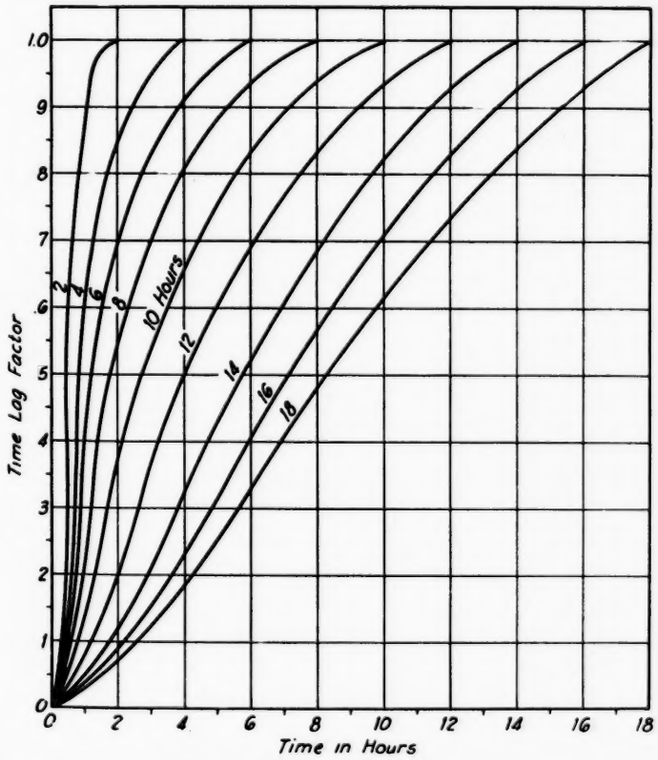


Fig. 4. Time lag factor curves.

TABLE 11—TIME LAG IN TRANSMISSION OF SOLAR RADIATION THROUGH WALLS AND ROOFS

Type and Thickness of Wall or Roof	Time Lag, Hours
Roofs	
Horizontal, 4-inch gypsum, waterproofing, slag finish	4½
Horizontal, 6-inch concrete, waterproofing, slag finish	5
One-inch concrete, 4-inch cinders, 1½-inch concrete, waterproofing, smooth black finish	8
Walls	
Wood siding, 1-inch sheathing, 2 × 4-inch studs, (studding space filled with insulation) lath and plaster	5
Brick, sheathing, 2 × 4-inch studs, lath and plaster	7
Brick, 8-inch tile and plaster	10½
Brick (13-inch), plastered	12
Brick (9-inch), 3¾-inch tile, 5¼-inch air space, 3¾-inch tile and 1¼-inch plaster	16

For example, assume a west wall at 40 deg N. latitude for which the time lag is eight hours and the heat gain from sun effect is desired at 3 p.m.

Two p.m. is one hour before 3 p.m. Enter Fig. 4 at the bottom and follow one-hour line vertically to intersection of eight-hour time lag curve and read time lag factor, L_t, equal to 0.2 on the left margin. Repeat for 1 p.m., 12 noon etc., until the maximum product is obtained. The results are:

$$\text{For 2 p.m., } I \times L_t = 153 \times 0.2 = 30$$
$$\text{For 1 p.m., } I \times L_t = 94 \times 0.54 = 50$$
$$\text{For 12 noon, } I \times L_t = 26 \times 0.7 = 18$$

This summary shows that the maximum is 50 at 1 p.m. Therefore, the peak heat gain at 3 p.m. will be obtained by multiplying

$$F \times a \times I \times L_t = \text{Btu per hr (sq ft) of net wall surface}$$

where

a is obtained from Table 8
I is obtained from Table 7
L_t is obtained from Fig. 4

Calculation of the sun effect through the wall using time lag and omitting it will point out the reduction in heat gain that occurs by allowing for time lag. The improvement is not as great as might be expected, however, because allowing for time lag increases the heat gain through walls that are not exposed to the sun in the afternoon but were exposed earlier in the day.

The sun effect heat gain through windows and doors may be assumed as instantaneous. Therefore, no allowance for time lag is ever made.

Since time lag is considered only in connection with sun effect, time lag can be neglected in connection with inside walls which are not exposed to the sun.

Summary of Method of Calculation

A summary of the steps in a complete calculation is given in Table 12. Reference is made opposite each item to the source of information.

Heat Gain Calculation Sheet

The calculation sheet illustrated in Fig. 5 is arranged so that the steps which have been described in detail, and summarized, follow in logical order, and so that spaces are provided for all the calculations. To illustrate the use of the Heat Gain Calculation Sheet each step in determining the heat gain of the room sketched in Fig. 6 is described.

The data necessary for making the calculations are: Latitude—35 deg. N.; number of floors in house—1; outside walls—brick veneer; roof—wood shingles; basement—cool; average number of persons in room —3; U_w for outside wall = 0.24; U_w for inside wall = 0.34; U_c = 0.28; U_r = 0.48; U_g for window = 1.1; U_g for inside door = 0.75; design maximum outdoor dry bulb temperature = 96F; design maximum outdoor wet bulb temperature = 77F; mean daily temperature range for July = 22F; design indoor dry bulb temperature = 78F; design indoor wet bulb temperature = 65F; design indoor absolute humidity = 72 gr per lb.

The time of day for which calculations are to be made is found from Table 9. From Fig. 6 it will be seen that the room has exposures on the west, south and east sides. There is a greater east exposure than west. Calculations both with and without awnings should be made for 2 p.m.

The design outdoor temperature for 2 p.m. is (see description Table 10)

$$96 - (0 \times 22) = 96$$

The design temperature differences across all walls are:

Outside wall — (96 — 78) = 18F
Inside wall — 15F
Ceiling (see Table 1) — 18F
Floor (the basement is cool, and hence the temperature differential is assumed zero).

These temperature differences are inserted in their proper spaces in Column V, and the outdoor-indoor difference is also inserted in the second line of the lower section opposite "Ventilation air."

The design humidity difference, 37 gr per lb, is inserted in the first line under "Latent Heat Gain."

The required ventilation air is actual infiltration from Table 4.

The infiltration rate is, from Table 5, approximately 1½ changes per hour, which corresponds to

$$\frac{3290 \times 1.5}{60} = 80 \text{ cfm}$$

In this case the infiltration rate is greater than the required amount of ventilation, hence in the space opposite "Ventilation air" in both the sensible heat gain and latent heat gain sections, insert 80.

Glass and wall areas are calculated in Column III from the data in Column II.

The transmission coefficients are written in the spaces provided in Column IV. U_w for the ceiling and roof combined is calculated in the column "Remarks," according to the formula given in Table 1 for an unventilated attic.

Conduction from temperature difference is calculated as the product of values in Columns III, IV and V, and the values inserted in Column VI.

From U_w in Column IV, F = 0.04 for all outside walls, and 0.057 for the roof. From Table 8, the absorption coefficient, A, is 0.7 for walls and roof. The values of I are obtained from Table 7 for 2 p.m. (35 deg. N. latitude).

East wall	24
South wall	83
West wall	152
Horizontal surface (roof)	245

Finally, the values in Column VII are multiplied to obtain R_w in Column VIII.

The glass radiation coefficients, R_g, are read directly for 2 p.m. from Table 7 and inserted in Column VIII.

The sun effect is obtained by multiplying the R_g in Column VIII by the glass area in Column III and the R_w in Column VIII by the net wall area in Column III.

The total conduction heat gain is the sum of Column VI, 6380 Btu per hr.

The sensible heat from ventilation air is calculated in the second line of the lower section of the calculation sheet. This result is 1560 Btu per hr.

The total sun effect (sum of Column IX = 8,700) is inserted in the third line opposite Sun Effect.

Sensible heat from people is calculated in the fourth line of the lower section. Assuming that the occupants are not engaged in any appreciable degree of exercise, the result is 3 × 220 = 660 Btu per hr.

In this case there is no sensible heat from electrical appliances or other miscellaneous sources.

The total sensible heat gain without window shading is the sum of Conduction, Ventilation Air, Sun Effect and People (17,300 Btu per hr).

The reduction of sun effect by using awnings is the product of the total glass sun effect (4,435 = 530 + 2,185 + 1,720, from Column IX) and a shading factor for awnings. The result is a reduction of 3,200 Btu per hr by using plain canvas awnings.

The total sensible heat gain with awnings is then 14,100 Btu per hr.

The latent heat gain from ventilation air is calculated in the first line under Latent Heat Gain, with a result of 2,000 Btu per hr.

The latent heat gain from people is calculated from the value 180 given under explanations for "Average Condition," with a result 3 × 180 = 540 Btu per hr.

In this case there are no other sources of latent heat gain.

The total latent heat gain is the sum of Ventilation Air and People (2,540 Btu per hr).

The total heat gain, with awnings, is the sum of the total corresponding sensible heat gain and the total latent heat gain (total heat gain, with awnings = 14,100 + 2,540 = 16,640 Btu per hr.

The sun effect on walls and roof total 90 + 230 + 335 + 3,590 = 4,265 Btu per hr, or 4,265 ÷ 16,640 × 100 = 25 per cent of the total load so that time lag may be neglected.

Unusual Sources of Heat Gain

Up to this point the discussion has centered on the most common sources of heat gain which are encount-

ered in air conditioning work. It is seldom that heat sources other than these are encountered, but occasionally an unusual case arises and it is in order to sound a warning about these.

One source of additional heat gain is sometimes found in the duct systems which are used with central plant air conditioning systems. These ducts are sometimes carried through hot attic spaces, and since the

Name ~~~ Room used for DINING ROOM Job No. ~~~

Street ~~~ Room No. Floor Sheet No. 1

City 35 DEG. N. LATITUDE Calculated by Date

DESIGN CONDITIONS

Outdoor: D.B. Temp. 96 °F. W.B. Temp. 77 °F. Abs. Hum. 109 Gr. per lb. Calculations are for 2 ~~a.m.~~ p.m.

Indoor: D.B. Temp. 78 °F. W.B. Temp. 65 °F. Abs. Hum. 72 Gr. per lb. Volume of room 3290 cu.ft.

Temp. Diff. 18 °F. Hum. Diff. 37 Gr. per lb. Infiltration rate $\frac{3290 \times 1.5}{60} = 80$ cfm.

Ventilation Air, Required Amount USE ACTUAL INFILTRATION cfm. (at indoor design conditions).

SENSIBLE HEAT GAIN

Column	I	II	III	IV	V	VI	VII			VIII	IX	
SIDE	Direction Wall Faces	Wall or Partition Dimensions / Windows and Doors, Number and Dimensions	Gross Wall Area, Sq. Ft. / Total Glass Area, Sq. Ft. / Net Wall Area, Sq. Ft.	— / U_g / U_w	Design D.B. Temp. Diff. °F.	Conduction from Temp. Diff. Btu. per hr. III×IV×V	Calculation of R_w for VIII $R_w = F \times A \times I$ F	A	I	— / R_g / R_w	Sun Effect Btu. per hr. III×VIII	REMARKS
1st	E	21'-6" × 9'	194	—		—	—	—	—	—	—	
		2W 3'×5'-6"	33	1.1	18	650	—	—	—	16	530	OUTSIDE WALL
			161	0.24		700	.04	.7	24	.67	110	
2nd	S	17'×9'	153	—		—	—	—	—	—	—	
		2W 3'×5'-6"	52	1.1	18	1030	—	—	—	42	2185	OUTSIDE WALL
		1D 2'-8"×7'	101	0.24		440	.04	.7	83	2.3	230	
3rd	W	10'-6" × 9'	95	—		—	—	—	—	—	—	
		1W 3'×5'-6"	17	1.1	18	340	—	—	—	101	1720	OUTSIDE WALL
			78	0.24		340	.04	.7	152	4.3	335	
4th	W	11'×9'	99	—		—	—	—	—	—	—	
		1D 4'×7'	28	0.75	15	310	—	—	—			INSIDE WALL
			71	0.34		360						
5th	N	17'×9'	153	—		—	—	—	—	—	—	
		1D 2'-6"×7'	18	0.75	15	200	—	—	—			INSIDE WALL
			135	0.34		690						
Ceiling		21'-6"×17'	366	—		—	—	—	—	—	—	
			—		18	—	—	—	—	—	—	$U_w = \frac{0.28}{14.7 \times \frac{0.28}{0.98}} = 0.20$
			366	0.2		1320	.057	.7	245	9.8	3590	
Floor		NEGLECT - COOL BASEMENT		—		—	—	—	—	—	—	
						—						
						—						

Total Conduction 6380 Total Sun Effect 8700

Conduction: Sum of Col. VI	= 6380	Btu. per hr.
Ventilation air: Cfm. (80) × Temp. diff. (18) × 1.08	= 1560	
Sun effect: Sum of Col. IX	= 8700	
People: No. of persons in room (3) × 220	= 660	
Electrical appliances: Watts (0) × 3.4	= —	
Other sources: Btu. per hr. 0	= —	
TOTAL SENSIBLE HEAT GAIN (Without window shading)	= 17300	Btu. per hr.
Saving from awnings or shades: Total glass sun effect (4435) × shading factor (.72)	= 3200	
TOTAL SENSIBLE HEAT GAIN (With window shading)	= 14100	Btu. per hr.

LATENT HEAT GAIN

Ventilation air: Cfm. (80) × Hum. diff. (37) × 0.68	= 2000	Btu. per hr.
People: No. of persons in room (3) × Btu. per hr. per person (180)	= 540	
Other sources: Lb. of water per hr. (0) × 1060	= —	
TOTAL LATENT HEAT GAIN	= 2540	Btu. per hr.
TOTAL SENSIBLE HEAT GAIN	= 14100	
TOTAL HEAT GAIN	= 16640	Btu. per hr.

EXPLANATIONS

U—Transmission Coefficient.
R—Solar Radiation Coefficient.
I—Intensity of Sun.
A—Absorption Coefficient.
F—Radiation Factor.
w (subscript)—Wall.
g (subscript)—Glass

See Engineering Data Book for coefficients and factors.

Heat from People, Btu. per. hr. per person.
Average Condition (at rest, eating, office work, etc.)
Sensible.....220
Latent.....180
Medium rate of exertion (restaurant waiters, average dancing, etc.)
Sensible.....330
Latent.....450

SE-40-24 Formerly 740-1 (9-45)

Fig. 5. General Electric heat gain calculation sheet. Problem used is related to Fig. 6.

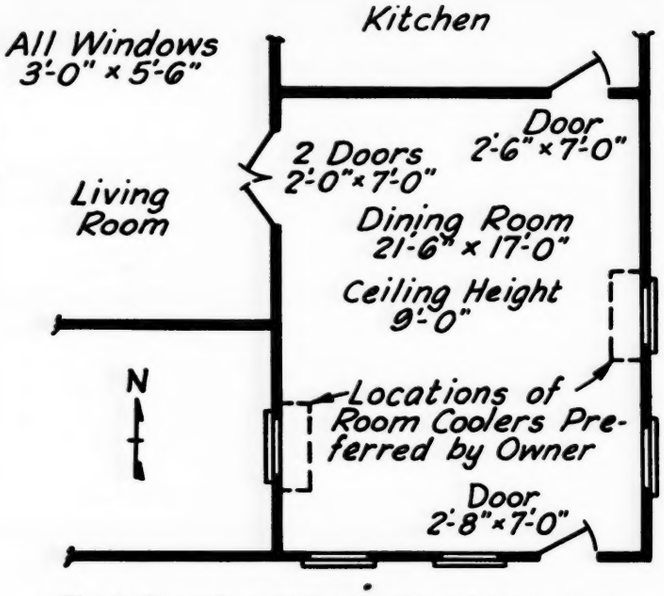


Fig. 6. Room for which heat gain calculations are shown in Fig. 5.

air within the ducts is always cooler than the air surrounding the ducts there may be a considerable transmission of heat through the duct walls. In one case where 400 feet of return duct work were carried through a ventilated attic, the calculated heat gain in the ducts alone was approximately 50,000 Btu per hr.

The heat gain in such cases must be calculated from fundamental data. In general, the heat transmission by convection and conduction through the duct wall may be found from a formula such as the following:

Heat gain, Btu per hr = (area of duct surface $\times$ U_d $\times$ (mean temperature difference between air inside duct and air outside duct, F).

where U_d = overall heat transmission coefficient from air inside duct to air outside duct, in Btu per sq ft per F per hr.

The overall transmission coefficient, U_d, in cases where the ducts are uninsulated, is derived from the two film transfer coefficients which apply to either side of the metal surface.

Average values for air conditioning applications to use in this calculation may be taken as 4.0 Btu per hr (sq ft) (F) for the inside film coefficient, and 1.7 Btu per hr (sq ft) (F) for the outside surface coefficient. The value 1.7 has been selected to include the effect of radiation as well as convection to the outside duct surface. Under these conditions the overall transmission coefficient, U_d, becomes (for an uninsulated duct):

$$U_d = \frac{1}{\frac{1}{1.7} + \frac{1}{4}} = 1.2$$

The formula for heat gain in an uninsulated duct then becomes approximately

Heat gain Btu per hr = (area of duct surface, sq ft) $\times$ 1.2 $\times$ (mean temperature difference between air inside duct and air outside duct, F).

In the case of insulated ducts it is necessary to correct the value of U_d to include the effect of the insulation. Most insulations have a conductivity of approximately 0.3 Btu per hr (sq ft) (F) (inch). The formula for U_d for an insulated duct would then become

$$U_d \text{ (insulated duct)} = \frac{1}{\frac{1}{1.2} + \frac{t}{0.3}}$$

where t = thickness of insulation, in inches.

Another unusual source of heat gain arises when rooms or furred spaces are used as plenum chambers. Sometimes the walls of such chambers adjoin very hot spaces, like kitchens, or attics, for example. Sometimes the chamber walls are outside walls, exposed to the sun. When such cases are encountered, it is necessary to calculate the heat gain of the plenum chamber as well as the heat gain of the air conditioned spaces.

A third case arises in places where steam pipes are hidden in walls and carry steam even in summer time. This might happen in hospitals or various industrial plants, for example.

TABLE 12—STEPS IN HEAT GAIN CALCULATION

Steps in Consecutive Order	Reference
Determine outdoor and indoor design conditions, as follows.	
Outdoor dry bulb temperature	Table 2
Outdoor wet bulb temperature	Table 2
Outdoor absolute humidity	Psychrometric chart
Indoor dry bulb temperature	Table 3
Indoor wet bulb temperature	Table 3
Indoor absolute humidity	Psychrometric chart
Determine design temperature difference	
Determine design humidity difference	
Determine required amount of ventilation air	Table 4 or 5
Calculate volume of room	
Calculate infiltration rate	Table 5
Determine time of day to use in calculations	Table 9
Calculate glass and net walls areas	
Determine heat transmission coefficients U_w and U_g for all walls, partitions, etc.	Reference books
Fill in design D.B. temperature difference for inside and outside walls	Fig. 5
Calculate conduction produced by temperature differences	Fig. 5
Calculate wall radiation coefficients, R_w, for time of day when total load is maximum	Fig. 5
Determine glass radiation coefficients, R_g, for time of day when total load is maximum	Table 7
Calculate total sun effect on walls and windows	Fig. 5
Fill in value of total sun effect	Fig. 5
Calculate sensible heat gain from people	Fig. 5 and reference books
Calculate sensible heat gain from electrical appliances	Fig. 5 and Table 6
Calculate sensible heat gain from other sources	
Calculate total sensible heat gain (without window shading)	Fig. 5
Calculate reduction in sun effect secured through window shading	Fig. 5
Calculate total sensible heat gain (with window shading)	Fig. 5
Calculate latent heat gain from ventilation air	Fig. 5
Calculate latent heat gain from people	Fig. 5
Calculate latent heat gain from other sources	Fig. 5
Calculate total latent heat gain	Fig. 5
Calculate total heat gain	Fig. 5
Calculate effect of time lag, if important	Fig. 5

Bind in.

LAMP REQUIREMENTS FOR ULTRAVIOLET IRRADIATION

It is frequently impractical to provide sanitary ventilation by actual air changes, so that equivalent ventilation, by use of germicidal lamps, may be employed. Ultraviolet irradiation by means of such lamps can be accomplished by lamps suspended in the room, or by lamps in the duct leading to the room, or by a combination of the two.

The following data show the method of calculation to determine the number of lamps required, and are presented through the cooperation of Dr. William T. Anderson, Director of Research, Hanovia Chemical & Manufacturing Company.

Room Lamps

Survey. The factors which must be determined when an installation is surveyed for room lamps are: volume of the room; ray length factor for distance between lamps and ceiling; ceiling height; maximum number of persons occupying the room, and relative humidity.

TABLE 1. VARIATION OF RADIATION WITH RELATIVE HUMIDITY

Relative Humidity, Per cent	Radiation in Watts per Sq Ft per hour (W)
50 or less	35×10^{-6}
55	41×10^{-6}
60	69×10^{-6}
65	179×10^{-6}

Calculating Lamp Requirements. For use in computing lamp requirements, the following four formulas are given:

$$H = \frac{N \times 30{,}000}{C} \qquad (1)$$

where H = lethes per hour (a lethe is the sanitary equivalent of one air change, outside air, per hour),

N = maximum occupancy of room in number of persons, and

C = volume of room, cubic feet.

$$I_t = H \times W \qquad (2)$$

where I_t = theoretical average intensity of radiation at 2537 angstroms wave length in the room, in watt-hours per square foot,

W = energy required per lethe, in watt-hours per square foot. This varies with the humidity as shown in Table 1; ordinarily use the value for 50% relative humidity.

$$E_t = I_t \times C^{2/3} \qquad (3)$$

where E_t = total radiation at 2537 angstroms, or total lamp output, in watt-hours,

C = volume of room in cubic feet. Values of $C^{2/3}$ for various values of C appear in Table 2.

$$L = \frac{E_t \times K}{B \times R} \qquad (4)$$

where L = number of lamps required,

K = an efficiency factor depending on the lamp mounting, and having a value of 5 for suspended lamps and 7 for wall-mounted lamps,

B = bactericidal lamp rating in watts at 2537 angstroms, and

R = ray length factor. See Table 3.

Example. A schoolroom with 41 persons maximum occupancy and $26 \times 32 \times 13$ feet is to be equipped with bactericidal lamps, each of 6.2 watts at 2537 angstroms, centrally suspended 7 feet from the floor. Only the upper air is to be irradiated, and the room relative humidity is 50%. How many lamps are required?

Solution. $C = 26 \times 32 \times 13 = 10{,}800$ cu ft, so that by equation (1):

$$H = \frac{41 \times 30{,}000}{10{,}800} = 114 \text{ lethes per hour.}$$

The value of W at 50% relative humidity is, from Table 1, 35×10^{-6} watts per square foot per hour, so that by equation (2),

$I_t = 114 \times 35 \times 10^{-6} = 0.004$ watts per sq ft. From Table 2, for $C = 10{,}000$, $C^{2/3} = 464$, and for $C = 11{,}000$, $C^{2/3} = 495$; therefore, $C = 10{,}800$, a value of $C^{2/3} = 490$ is assumed and by equation (3)

$$E_t = 0.004 \times 490 = 1.96 \text{ watts}$$

TABLE 2. VALUES OF $C^{2/3}$

C	$C^{2/3}$	C	$C^{2/3}$	C	$C^{2/3}$
1000	100	10,000	464	25,000	855
2000	158	11,000	495	30,000	965
3000	208	12,000	524	35,000	1070
4000	252	13,000	553	40,000	1170
5000	292	14,000	581	45,000	1265
6000	330	15,000	608	50,000	1357
7000	366	17,500	674	60,000	1533
8000	400	20,000	737	70,000	1698
9000	433	22,500	797	80,000	1857

Since the lamps are to be suspended, K = 5. The lamp rating *B* was given as 6.2. For a 7-foot mounting height, and with a 13-foot ceiling (given), the distance from lamp to ceiling is 13 — 7 = 6 feet; from Table 3 for 6-foot distance, *R* = 0.91, so that by equation (4)

$$L = \frac{1.96 \times 5}{6.2 \times 0.91} = 1.74 \text{ lamps.}$$

The nearest larger whole number is 2 (lamps required).

Duct Lamps

Survey. The factors to be determined for an installation of ultraviolet lamps located within a room are: dimensions and volume of the section of duct to be irradiated; volume of room; maximum occupancy of room served by duct; volume of air flow through duct; ray length factor, and relative humidity.

Calculating Lamp Requirements. Four formulas are required, of which the first is as given in equation (1) in the foregoing. The others are

$$H_{max} = \frac{A}{C} \qquad (5)$$

where H_{max} = maximum lethes per hour available from duct,
A = air flow through duct, cubic feet per hour,
C = room volume, cubic feet

$$I_d = H_{max} \times W \qquad (6)$$

where I_d = theoretical average intensity of radiation at 2537 angstroms wave length in duct and W is as given in equation (2).

$$E_d = \frac{I_d A}{V^{1/3}} \qquad (7)$$

where E_d = total radiation at 2537 angstroms, or total lamp output, in watts,
V = volume of irradiated section of duct in cubic feet. A is as in equation (5).

$$L = \frac{E_d \times K}{B \times R} \qquad (8)$$

where L = number of lamps required,
K = efficiency factor = 4 for duct mounting
R = ray length factor based on one-half the minor duct dimension. See Table 3.
B is as given in equation (4)

Example. A 4000 cu ft room is supplied by a 2 ft × 2 ft duct of which a 6 foot length is to be irradiated by 6.2 watt lamps as rated at 2537 angstroms. Air supplied through the duct is 30,000 cu ft per hour, the relative humidity is 50%, and the occupancy of the room is 10 persons. What are the lamp requirements?

Solution. By equation (1)

$$H = \frac{10 \times 30{,}000}{4{,}000} = 75 \text{ lethes per hour}$$

The maximum lethes per hour obtainable from the duct are, by equation (5)

$$H_{max} = \frac{30{,}000}{4{,}000} = 7.5 \text{ lethes.}$$

It is seen that 75 — 7.5 = 67.5 lethes per hour must be provided by room lamps, since the duct is capable of supplying only a portion of the load. The room lamps would be calculated as shown previously.

The value of *W* in equation (6) at 50% relative humidity is, from Table 1, 35 × 10^{-6}, so that

$I_d = 7.5 \times 35 \times 10^{-6} = 0.00026$ watts per sq ft per hour.

The volume of irradiated duct *V* is 2 × 2 × 6 = 24 cu ft, and $V^{1/3}$, the cube root of *V* is the cube root of 24 or about 2.9, so that, with equation (7)

$$E_d = \frac{0.00026 \times 30{,}000}{2.9} = 2.7$$

The minor duct dimension is 2 ft, and one-half of this is 1. From Table 3, for 1 ft, R = 0.33. B is given as 6.2, so that from equation (8)

$$L = \frac{2.7 \times 4}{6.2 \times 0.33} = 5.3 \text{ lamps, or 6 lamps in the duct.}$$

It is important to consider that disinfection results from the product of intensity of ultraviolet and a time factor. If one is increased, the other may be decreased proportionately.

TABLE 3. RAY LENGTH FACTOR *R*

Distance in Feet from Lamp to Ceiling or from Lamp to Nearest Side of Duct	Ray Length Factor *R*
0.25	0.10
0.50	0.20
0.75	0.27
1.0	0.33
1.5	0.45
2.0	0.54
2.5	.....
3.0	0.68
4.0	0.78
5.0	0.86
6.0	0.91
7.0	0.94
8.0	0.97
9.0	0.98
10.0	0.99
11 and over	1.00

Effect of Hot Spells on Mortality Measured in New York

IN the first intensive study of its kind ever conducted by the city, the Department of Health of the City of New York has drawn preliminary conclusions on the summer weather conditions that make New York's death rate climb sharply.

At the direction of Dr. Harry S. Mustard, Health Commissioner, the survey was begun last September by the agency's bureau of records and statistics to discover the exact relation between heat waves and death rates. City weather charts and death records covering the May to September periods for the last twenty-five years were studied.

Carl L. Erhardt, bureau director, reports that the study thus far showed that rises in mean temperatures above 80F for four consecutive days had been associated with increases of 20% or more above normal in summer weekly death rates since 1925 in all instances except one. When the mean temperature exceeded 85F for at least two of these four days, the increase in the city's weekly death rate was much sharper—40% or more above normal.

Mr. Erhardt added that mean temperatures above 80F for less than three days had produced less abrupt increases in death rates.

Made Mortality Graphs

Mr. Erhardt's staff graphed the weekly mortality rates for the last twenty-five summers and charted the maximum and mean temperatures for the same periods. The cycles on both graphs generally followed the same pattern, except for the weeks ended June 8, 1940 and July 12, 1941.

Although the mean temperatures for the latter week were above 80F for five consecutive days, the death rate for the week did not rise above normal. During the week ended June 8, 1940, the death rate rose 30% above the normal, but there had been only one day of temperature above 90F for the seven-day period. And on none of these days did the mean temperature exceed 80F.

These puzzling discrepancies in the study have led the statisticians to delve deeper into the records for other factors besides high temperature that may cause increased death rates in summer months. The effect of hours of sunshine, of rainfall, of changes in barometric pressure and of wind velocity are other meteorological conditions that must be taken into account in this study.

The bureau has discounted relative humidity as a factor because it has been found that during the weeks when the death rates were highest, with steadily high temperatures as well, relative humidity was not excessive.

The most serious increases in the mortality rate over the last twenty-five summers, the study revealed, occurred during the following periods:

Week ended June 13, 1925, when the death rate was 60% above normal. The mean temperature ranged from 80 to 89F on four consecutive days.

Week ended July 18, 1936, when the death rate was 50% above normal. The mean temperature on five consecutive days fell between 80 and 89F.

Week ended July 18, 1937, when the death rate was 70% above normal. The mean temperature was again between 80 and 89F for five consecutive days.

Week ended September 3, 1948, when the death rate doubled. Mean temperature was between 80 and 89F on three days and over 90F on three days, making a consecutive period of six days of excessive mean temperatures.

Eleven other summer weeks in the twenty-five-year period showed death rates which went above the normal by 30 to 40%. Seven weeks showed mortality figures 10 to 20% above normal. During each of these eighteen weeks, the study found, temperatures fell into the excessive category for as many as six consecutive days during some of the weeks.

Importance of the Study

"This information," Mr. Erhardt commented, "is important scientifically to determine precisely what the danger degree of heat is, and what other factors, if any, are related, so that the public may be warned about the risk involved during such periods. Other analyses of deaths during protracted hot spells have shown that the very young, the very old and the chronically ill are particularly affected. It is important, therefore, that extraordinary precautions be taken to keep these groups comfortable when the heat graph goes up on a successive number of days."

Mr. Erhardt added that so far this summer the death rate had not risen 10% above the normal for any one week. This, he explained, checked with the survey's tentative conclusions because there had not been a mean temperature this year above 80F for four consecutive days.

Caution

The bureau will continue to watch the weather charts and death records the rest of the summer and report its final survey results when the effects of meteorological conditions other than temperature have been fully investigated. With the usual caution of a statistician Mr. Erhardt emphasized that the present facts should not be interpreted as final conclusions in view of the investigations yet to be made.

Air Conditioning Design Conditions for Various Industries

NATHAN N. WOLPERT
Associate Editor, Heating and Ventilating

Part 4, the concluding section, of a group of articles on design conditions considered satisfactory for air conditioned rooms in industry. In all, thirty manufacturing processes or industries are considered in the series, including an article on air conditions for textiles. Data are presented on air control for key processes in many industrial applications.

PARTICULAR attention is directed to the extensive data on pharmaceuticals that is presented in this article. This information is considered outstanding because most of the manufacturers of pharmaceutical and bacteriological products are reserved regarding air conditions that they maintain in key rooms or processes. Because of the conditions that must be maintained in many of the pharmaceutical processes, considerable attention is given to means for obtaining dust-free, sterile air.

This article, which is the last of a group of four, includes data on some of the metal industries.

Automobiles

Controlled air conditions for various rooms in the manufacture of automobiles are listed in Table 15. These conditions are employed by a major automobile manufacturer.

In offices, drafting rooms and matching rooms where employes spend the entire day, the dry bulb temperature is maintained higher during the summer months to avoid the possible health hazard resulting when one moves from conditioned zones to high temperatures that may prevail outdoors.

TABLE 15—AIR CONDITIONS FOR AUTOMOBILE MANUFACTURE

Room or Process	Temperature, F	Relative Humidity, %
Engineering offices and drafting rooms		
Summer	80	40
Winter	75	35
Spectrographic analysis room		
Summer	80	50
Winter	75	45
Gear matching and special assembly rooms		
Summer	80	40
Winter	75	35
Gasket storage		
All seasons	100	50
Cement and glue storage		
All seasons	65	40

Another manufacturer employs air conditioning for—

(1) Precision machining of parts.
(2) Accurate gaging and inspection.
(3) Manufacture and adjustment of precision gages.
(4) Processing of certain kinds of material which can be satisfactorily worked only if their moisture content and temperature are correct.

Conditions maintained for operations (1) and (2) are generally 75F and 45-50% R. H. For (3), conditions are generally 68F and 50% R. H., although at times 75F and 50% R. H. may be permitted. For (4), a great deal depends upon the operation. In one case, a year-round temperature of 65F and 10-15% R. H. is necessary.

For general manufacturing and assembly areas, no attempt is made to control conditions during hot weather. An ample supply of fresh air and air motion are relied upon to provide personal comfort.

The concentration of dust and fumes is continually checked to see that they are maintained below the maximum permitted by the State Industrial Health Department.

Gage Room

The last World War brought out the importance of maintaining standard air conditions in gage rooms so that parts manufactured in several plants could be assembled to form one machine or unit. Most of the industrial nations have adopted 68F or 20C as the standard temperature for gage rooms. This figure was selected because it is a reasonable room temperature that is fairly easy to maintain; it is expressed as a whole number on whichever of the two temperature scales that is used; and it was adopted in some plants turning out metric gages.

This standard condition for gages has made possible a very close tolerance. For example, during World War I, 0.002 inch was considered a close tolerance; during World War II, a tolerance of 0.0002 inch was not unusual and was required on many government specifications.

Relative humidity is maintained at 40-50%, and dust control is necessary for a quality product.

Laminated Glass

In the manufacture of laminated or safety glass, a sheet of plastic is used as a filler or binder between two sheets of glass. It is important that the two pieces of glass be dust-free to produce a clear product, and that the plastic binder be properly dried to remove intercellular moisture.

TABLE 16—AIR CONDITIONS FOR LAMINATED GLASS MANUFACTURE

Room or Process	Temperature, F	Relative Humidity, %
Cutting room	65	15
Vinyl inspection and laminating room	55	15
Vinyl drying ovens	142	5

The plastic (vinyl resin) is slightly hygroscopic under humid conditions so that it may take up enough moisture to prevent it serving as a satisfactory binder for the glass sheets. At ordinary room temperature it is limp and tacky but at temperatures below 65F it becomes fairly stiff and it loses its stickiness.

Air conditions to be maintained in this industry are listed in Table 16.

All air supplied to the dehumidifier serving the cutting room is double filtered to protect the alumina beds from contamination.

Glass coolers on each assembly conveyor cool the glass from 110F, the temperature as it leaves the washers, to 60F as it enters the laminating room. It is important to keep the air in the cooler at low humidity to pick up any surface moisture that may remain on the glass from the washer, and to prevent condensation on the cooling coils.

Machinery

Controlled air conditions are important in the assembly rooms and in some machining operations. Machining tolerances and finishes of parts for precision gear drives are held to such specifications that assembly under constant temperature conditions is essential. Dust counts are held down fairly well in the room reserved for assembly and run-in of compressors.

TABLE 17—AIR CONDITIONS MAINTAINED IN MACHINE ASSEMBLY

Room or Process	Temperature, F	Relative Humidity, %
Assembly of precision gear drives	72	42-50
Assembly and run-in of air conditioning compressors	70-76	45-55

Pharmaceuticals

For the purpose of this article, grouped under the general term of pharmaceuticals is the manufacture of tablets, powders, capsules, ampules, toxines, and the maintenance of small animals for research work.

Conditions maintained in the various air conditioned departments are summarized in Table 18.

There is an especially fine powder that is sometimes used in filling ampules that requires that the relative humidity be maintained at 20%.

In the manufacture of uncoated pills, the temperature and relative humidity must be kept accurately controlled, for if the humidity is too high, the pills and tablets will not dry at the proper rate, and the powder

TABLE 18—AIR CONDITIONS FOR MANUFACTURE OF PHARMACEUTICALS

Room or Process	Temperature, F	Relative Humidity, %
Tablet compressing department	77	40
Powder storage and packing areas	75	35
Milling room	80	35
Ampule manufacturing department	80	35
Biological manufacturing department	80	35
Liver extract	80	10.5
Serums	74-78	50
Capsule storage	75	35-40
Animal room	80	40
Small animal room	75-78	47-48

that may be used as a surfacing will not adhere. If the humidity is too low, a hard outer layer may form before the inside of the pill has had a chance to dry thoroughly. After the pills have been bottled, then some of the retained moisture may be given out so that the pills may stick together. For coated pills, the molded centers are brought in contact with the liquid coating. The number of applications of the coating determines its final thickness.

Liver extract must be given special attention. Since liver is hygroscopic, a low humidity must be maintained after it is dried in a vacuum drying machine. Temperatures higher than 80F will cause the extract to deteriorate.

Drug manufacturers use large numbers of animals in testing and research work and also for production. While the animals used include guinea pigs, mice, rats, rabbits, monkeys, dogs, cats, chickens, hogs, horses and cows, the ones most frequently used in testing and research work are the smaller animals—mice, guinea pigs and rabbits. If test animals become sick and die while under observation as the result of weather conditions, it may mean tests will have to be repeated, and considerable time and money will be wasted.

In the research room, it is only necessary to maintain comfort conditions. However, in the microanalysis room, 80F and 50% R. H. are maintained throughout the year.

It is extremely important that sterile conditions be maintained in many of the pharmaceutical processes.

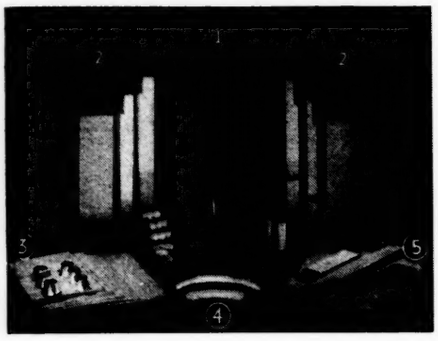
Watches

Air conditions are controlled in all rooms where watches are assembled. Temperature is held within the range of 72-75F, with a maximum relative humidity of 50%. Filters are used in the air circulating system to keep down dust. In the manufacturing divisions, the only rooms air conditioned are the mainspring department and the dial department. Temperature is held within a range of 72-75F mainly for comfort. However, in the manufacture of mainsprings, relative humidity is held to below 50% to control corrosion. By keeping down the relative humidity to 50% in the room where watch dials are lacquered and printed, many manufacturing troubles are eliminated. Previous to air conditioning, there were difficulties due to high humidity, particularly during the summer.

NEWS OF EQUIPMENT AND MATERIALS

Coleman Blend-Air

NAME—Coleman Blend-Air system of forced warm air heating.
PURPOSE—A packaged heating sys-

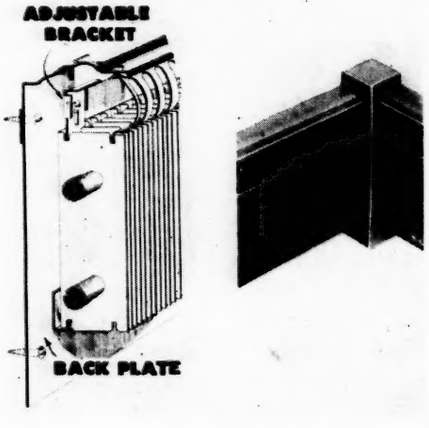


tem that employs 3½-inch round ducts.
FEATURES—System has three basic parts: a streamlined furnace; a system of factory-made supply ducts, and a series of blending units in which heated air from the furnace is mixed with air taken from room. Heated air from furnace is forced through small ducts at temperatures up to 195F. Entering the blender unit, the heated air passes through a nozzle and this action causes the room air to be sucked in through a grille into the blender to be mixed in the blender before being discharged through the air outlet located at the top of the blender. Blended air has a temperature of 135F and a velocity of 300 fpm. Each hot air run is said to deliver up to 10,000 Btu per hr. Only one short return duct is normally required to the furnace. Three types of blenders are available: (1) a telescoping model for new house construction which fits in between the studs of wall; (2) a unit that could be recessed in a wall; (3) a blender designed for use in existing homes to be set against the wall. Blenders can be adjusted to control the amount of mixed air entering the room. Entire system is operated by automatic room temperature controls. In photograph, (1) shows the oil-fired Coleman Modumatic furnace, (2) telescoping concealed type air blenders with attached grilles, (3) furnace bonnet assembly with flared supply duct connectors, (4) standard 3½-inch duct sections, and (5) parts of the return duct with grille.
LITERATURE AVAILABLE—Bulletin.
MADE BY—*The Coleman Company, Inc., Wichita, Kans.*

Item 287

Kritzer Baseboard System

NAME—Kritzer radiant baseboard unit.
PURPOSE—Space heating.
FEATURES—Radiant heating coil is constructed in 10-ft sections of 2 x 5½-inch aluminum fins mechanically bonded to two 3-inch center, copper supply tubes. No special fittings are required to join the ends of the copper tubes when installing the coils. As two sections are brought together, the supply ends in one section are belled and the ends of the other section are inserted into this bell. A solder ring heated and melted between the two ends completes the connection. An entire installation requires a steel back-plate, radiant coils, filler plates, and end posts which are interchangeable. Ornamental grilles are available where decoration is desired. Coils float on hanging clips attached to the back-plate to permit free circulation of cold air behind the coil.
INSTALLATION—Steel back-plate is placed against the wall on the finished floor level. Back-plate is then secured to the wall with either nails or screws. Where the wall is uneven, felt sealing strips are provided and these strips also prevent wall smudging. After the plate is secured to the wall, the coils are hung in place and the coil ends are soldered where necessary. Where required, end posts are used.



SIZES AND CAPACITIES—Overall size of baseboard with ornamental backplate, 2¼-inch deep, 9-inch wide.
MADE BY—*Kritzer Radiant Coils, Inc., 2901 Lawrence Ave., Chicago 25, Ill.*

Item 288

Dunham Convector

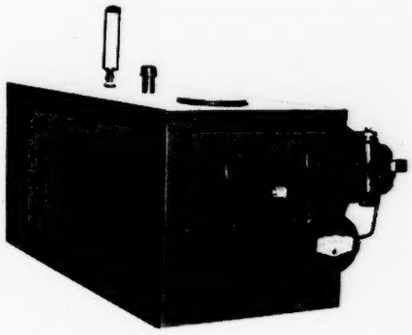
NAME—Dunham wall type convector.
PURPOSE—Space heating.
FEATURES—A low height convector with a sloping top is designed for use on either steam or forced hot

water installations and for working pressures up to 150 lb. A series of these units may be installed to form one long continuous convector and a sheet metal splice plate is available to conceal the pipe connections between each unit. Cabinet has a removable front which includes a horizontally slotted outlet grille. Unit is available with the choice of three types of 1¼ inch finned pipe heating elements, steel pipe with steel fins, steel pipe with aluminum fins, and copper pipe with aluminum fins. Brackets supporting the heating element are adjustable to vary the pitch or grade as much as 1½ inch in either direction.
SIZES AND CAPACITIES—Capacities are 13.9 to 53.2 EDR with one pound steam, 65F entering air. Made in lengths from 2 to 6 ft in 6-inch increments, with a height of 10⅜ inches and a width of 5½ inches.
MADE BY—*C. A. Dunham Co., 400 W. Madison St., Chicago 6, Ill.*

Item 289

Dewey-Shepard Boiler

NAME—Dewey-Shepard boiler.
PURPOSE—Horizontal type hot water boiler which can be fired by oil or gas.
FEATURES—Unit is considered to

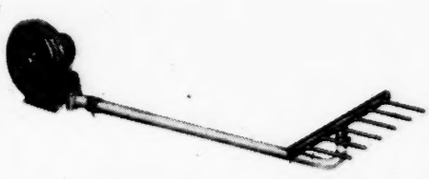


be adapted for baseboard, floor or ceiling panel hot water heating. Boiler is cylindrical and can be furnished with or without jacket. It can be fired interchangeably with Breese or gun type oil burner or gas-fired equipment. Boiler is manufactured under A.S.M.E. Code inspection and carries certificates of authorization and ratings with each unit. Oil-fired equipment is Underwriter approved and gas burners are A.G.A. approved, listed for use with natural, artificial, or L.P. gas.
SIZES AND CAPACITIES—Maximum capacity, 70,000 Btu per hr; recovery of 93 gal per hr at 87F. Outside jacket dimensions are 18 inches wide, 18 inches high and 39 inches long, with burner equipment projecting from this jacket.
MADE BY—*Dewey-Shepard Boiler Co., 1311 N. Capitol Ave., Indianapolis 4, Ind.*

Item 290

Pliojet System

NAME—Pliojet system for smoke abatement.
PURPOSE—Over-fire air system for smoke abatement through secondary air supply.

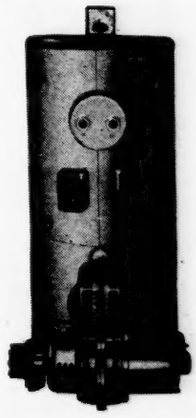


FEATURES—The system introduces secondary air through over-fire air jets which are arranged in sets. Ten standard sets make it possible to specify the arrangement that will provide adequate penetration at the maximum firing rate according to the individual needs of the firebox. The system consists of a turbo-blower, air valve, manifold, and properly selected jet nozzles. These nozzles are made of high temperature alloy or cast iron and are spaced from 6 to 11 inches apart. The blower is direct-connected to a standard motor and comes complete with magnetic starter. Manifolds have the couplings welded in place for attaching the jet nozzles. The secondary air that is supplied in the firebox provides the necessary oxygen and turbulence to obtain complete combustion of the gases.
LITERATURE AVAILABLE—Bulletin.
MADE BY—*Plibrico Jointless Firebrick Co., 1800 Kingsbury St., Chicago 14, Ill.*

Item 291

Delta Boiler-Burner

NAME—Delta LA-200 boiler-burner unit.
PURPOSE—For space heating with built-in tankless domestic hot water heater for small homes.
FEATURES—Shell is of ¼-inch steel. A.S.M.E. steel construction design is employed. In the boiler are 10-gauge tubes to provide 16 sq ft of heating surface. All connections are to the front of the boiler so that unit can be installed close to a wall. A flange mounted burner is designed to burn ¾ gal of oil per hr. Boiler is jacketed with 1½ inches of glass wool insulation. Unit is designed for oil or gas firing.



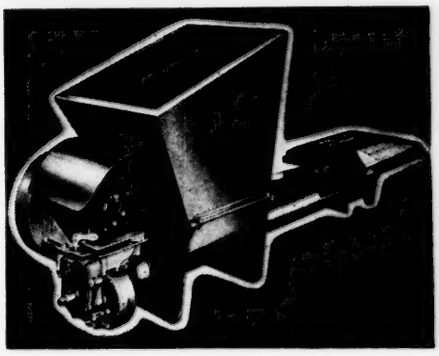
SIZES AND CAPACITIES — Output, 66,000 Btu per hr. Height of jacketed unit is 38 inches; diameter, 18 inches. Built-in tankless coil supplies 180 gal hot water per hour.
MADE BY—*Delta Heating Corp., 85-07 Northern Blvd., Jackson Heights, N. Y.*

Item 292

Brownell Stoker

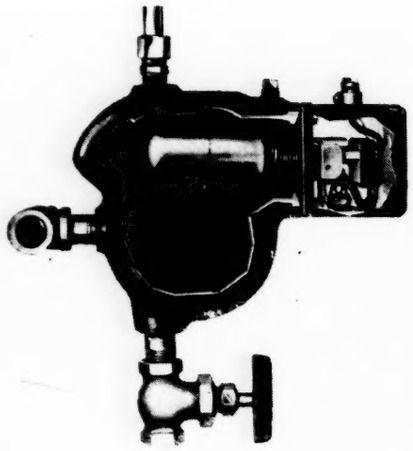
NAME—Brownell Type F stoker.
PURPOSE — Automatic burning of coal where heating requirements range from 1,800 to 10,400 sq ft of steam radiation.
FEATURES—Transmission is fully enclosed. Gears operate in a bath of oil while tubes and troughs bring lubrication to the bearings. Seals and leakproof packings prevent escape of oil. A V-belt delivers power from the motor. A worm on the shaft engages a phosphor bronze worm gear and the gears operate at slow speed. A channel runs from the bottom of the gear case to the oil level cup. If water accumulates, it forces a slug of oil out of the cup and this overflow of oil is a warning to the operator of the presence of water in the gear-case. Sheaves on motor and drive shafts provide three speeds of coal feed. Operating tension of the belt is easily adjusted by means of a set-screw. An air volume control is manually set to regulate the air intake. Fan and housing are of metal capacity to supply the proper amount of air for efficient fuel combustion. Fan is of multi-vane type and quiet in operation.
SIZES AND CAPACITIES — Available in seven sizes with coal burning rates from 75 to 312 lb per hr.
LITERATURE AVAILABLE — Bulletin S-41.
MADE BY—*The Brownell Company, 410 N. Findlay St., Dayton 1, Ohio.*

Item 293



Watts Switch

NAME—Watts mercury switch, low water cut-off, No. 89-A.
PURPOSE—Low water cut-off control for low pressure systems.

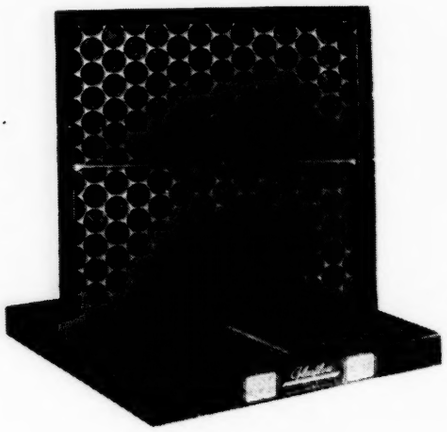


FEATURES — The surge-snubbing float chamber is said to assure clean-cut action on the switch during change of water levels regardless of turbulence in the boiler. Float is made of copper, built for long service. A mercury switch of heavy Pyrex glass is used for breaking the electrical circuit. Ample wiring space with barrier is included to protect the switch mechanism. Unit has a large mud space float chamber bowl.
SIZES AND CAPACITIES—Maximum steam pressure, 25 lb. Float chamber is 6½ inches high. Unit is 4 inches wide; 9 inches long; weighs 10 lb.
LITERATURE AVAILABLE—Folder.
MADE BY—*Watts Regulator Co., Lawrence, Mass.*

Item 294

Glasfloss Filter

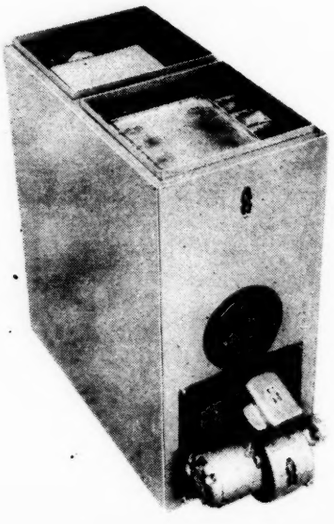
NAME—Glasfloss air filter.
PURPOSE—Air cleaning.
FEATURES—Long, fine glass fibers are assembled in a resilient mass and these fibers are bonded together with a high temperature bonding agent. The filter medium is then coated with a fire-resistant dust-catching adhesive. For special applications, filters can be supplied dry or with a reduced amount of adhesive. Filters are made for all types of central heating, ventilating, and air conditioning systems and are supplied in all standard sizes for use in home heating equipment and in commercial and industrial filter banks.
SIZES AND CAPACITIES—Supplied in 1- and 2-inch thicknesses in all standard sizes to handle 200 to 1,000 cfm.
LITERATURE AVAILABLE—Folder.
MADE BY — *Glasfloss Corporation, 155 E. 44th St., New York 15, N. Y.*



Item 295

Herco Furnace

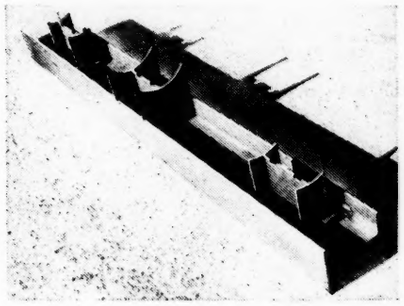
NAME — Herco stainless steel oil heating unit.
PURPOSE—Space heating.
FEATURES—Fire travel is through a stainless steel combustion chamber. A Sirocco blower unit is used. Motor is mounted on rubber for silent, vibration-free operation. The unit has a double-wall combustion chamber and heat economizer. Heat that is ordinarily lost through the chimney is captured by the economizer which serves to pre-heat the cold incoming air. The Herco burner has a firing rate of 0.65, 0.75 and 1.0 gph. Burner is operated by a 1/8 hp motor and the blower by a 1/6 hp motor. Glass type filters are used in the unit. Controls are either Mercoid or Minneapolis-Honeywell make. Fan and limit controls prevent overheating.
SIZES AND CAPACITIES — In three sizes from 65,000 to 110,000 Btu per hr.
LITERATURE AVAILABLE—Folder.
MADE BY—*Herco Oil Burner Corp., Lancaster, Pa.*



Item 296

Guide-Anchor-Guide

NAME — Guide-Anchor-Guide for expansion joints.
PURPOSE—Method for guiding and anchoring an expansion joint on long rigid members.
FEATURES—Anchor and guide connections are fabricated to fit and are welded to the pipe in the field. Connections are notched to eliminate any locked-up stresses in the piping due to weld shrinkage. Guide connectors are made with a base plate which slides on and between round bars. Guide results in alignment of working parts at and around the expansion joint. Guides are welded to the rigid member to permit free movement as the pipe expands or contracts.



SIZES AND CAPACITIES—Available to accommodate pipe diameters up to 36 inches.
LITERATURE AVAILABLE—Folder.
MADE BY — *Piping Specialties, Inc., 114 Liberty St., New York 6, N. Y.*

Item 297

Fairbanks-Morse Furnace

NAME—Fairbanks-Morse furnace, hi-boy model.

PURPOSE—Warm air space heating.

FEATURES—Company has announced a new line of gas- and oil-fired utility warm air furnaces styled as hi-boy models. Furnace is equipped with a high pressure atomizer-type oil burner powered by a 1/12-hp motor. Unit is provided with a 12-gage heat exchanger. Compactness of unit results from parallel spacing of plate-type flues within the furnace. Heat savings are also said to result from the use of corrugated plate flues. Corrosion-resistant alloy steel is used in flue construction.

SIZES AND CAPACITIES—Available in five sizes with bonnet outputs of 60,000, 80,000, 100,000, 150,000 and 180,000 Btu.

MADE BY—*Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.*

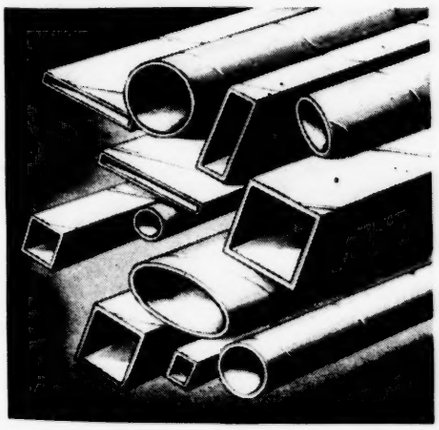
Item 298

Precision Tube

NAME—Precision asbestos tube.

PURPOSE—Heat resistant tubes of various shapes and sizes.

FEATURES—Tube is made by spirally winding especially prepared asbestos tape to predetermined sizes around a mandrel and it is then die formed into shape—either square, rectangular or oval cross section. Tubes, it is said, are unaffected by high temperatures. Tubes are made in any length with wall thicknesses from .01 inch up.



MADE BY—*Precision Paper Tube Co., 2045 W. Charleston St., Chicago 47, Ill.*

Item 299

Sahara Dehumidifier

NAME—Niagara Sahara dehumidifier.

PURPOSE—For reducing moisture in storage closets, stock rooms, and industrial areas.

FEATURES—Unit, which is plugged into any electric outlet, is provided with a hopper filled with calcium chloride crystals. A motor, connected to an axial flow fan, draws in air that flows over the crystals which extract the moisture. Moisture falls into a 2½ gal container. With normal use, makers claim that the unit requires recharging every month or so. A humidistat, which is optional extra equipment, may be installed to make the dehumidifier operate automatically. In a room kept at 75F and 75% relative humidity, makers claim unit is capable of lowering the relative humidity to 71% in two hours and to 66% in 10 hours.

SIZES AND CAPACITIES—15¾ inches x 15¾ inches x 41½ inches high.

LITERATURE AVAILABLE—Illustrated folder.

MADE BY—*Niagara Industrial Corp., 20 Vesey St., New York 7, N. Y.*

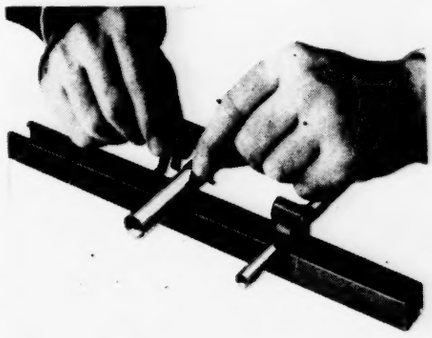
Item 300

Unistrut Tube Rack

NAME—Unistrut spring clamp for pipe rack.

PURPOSE—Support for tubes.

FEATURES—The steel channel is designed to accommodate tube sizes from 3/16 to 1 inch. O.D. Unit permits each tube to be fastened individually and permits expansion and contraction while holding the tube firmly in place. Rack can be mounted or hung in any position, and where two pieces of channel



are spot-welded back to back, tubing can be racked on both sides.

LITERATURE AVAILABLE — Bulletin TC-1.

MADE BY—*Unistrut Products Co., 1013 W. Washington St., Chicago, Ill.*

Item 301

Huron Baseboard

NAME—Huron baseboard radiation unit.

PURPOSE—Space heating.

FEATURES—Room systems are made up of individual units constructed of a series of aluminum fins with copper tube which runs through the center of the fins. Unit is designed to carry water at 180F. This element is said to provide a floor and ceiling temperature differential of 2F.

SIZES AND CAPACITIES—Units available in sections up to 12 ft in length. Fins are 7 inches high, 1¾ inches deep, and ⅜ inch wide. At 180F, output per 12 ft section, 7,140 Btu per hr.

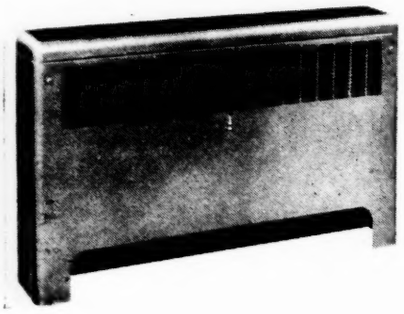
LITERATURE AVAILABLE—Illustrated folder.

MADE BY—*Huron Boiler Co., 12436 Van Dyke Ave., Detroit 34, Mich.*

Item 302

Airtherm Convector

NAME—Airtherm convector.
PURPOSE—Space heating.
FEATURES—New design of unit, covers changes in the convector en-



closure to provide greater strength and rigidity in the top section. The front panel design permits a simple and faster method of removing and assembling on the job. Spring operated damper is assembled onto the front panel. Damper can be adjusted between fully open and closed. Models are available in free standing or recessing, sloping top and regular front wall hung types, and flush recessing.
SIZES AND CAPACITIES—Available in 4-, 6-, 8-, and 10-inch depths and a wide range of lengths and heights.
LITERATURE AVAILABLE — Bulletin 701.
MADE BY—*Airtherm Manufacturing Co., St. Louis, Mo.*

Item 303

Crane Valve Operator

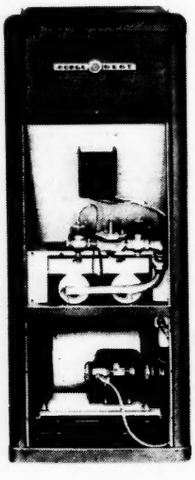
NAME—Crane fluid-motor operator for valves.
PURPOSE—A simplified fluid-motor operator for the operation of gate valves.
FEATURES—Motor unit, which consists of the motor, gear box and yoke, is bolted to the valve bonnet. The motor, which operates the valve stem through the gear box, can be served by water, oil or compressed air or gases. At pressures from 400 to 300 lb per sq in., operating mechanism is said to deliver a high starting torque; this is adjusted so as to deliver a greater torque for unseating the valve than for seating it. A handwheel is supplied for manual operation in the event of pressure failure. The motor derives its power from five flexible diaphragms mounted radially about an eccentric on the drive shaft. The operating fluid is admitted in rotational sequence to the five diaphragms which actuate pistons that transmit the thrust to the eccentric through a roller bearing mounted on the shaft. By means of a control, the operating fluid can be admitted through either of two entrance ports, controlling the direction of operation of the motor for opening and closing the valve.

SIZES AND CAPACITIES — Available in most types of Crane iron-body gate valves in sizes up to 30 inches.
MADE BY—*Crane Company, 836 So. Michigan Ave., Chicago 5, Ill.*

Item 304

Norge Furnace

NAME—Norge series F500 gas-fired winter air conditioner.
PURPOSE—Space heating.
FEATURES—Unit is approved by the A.G.A. for natural, mixed, manufactured and L.P. gases and is designed for use in utility room, closet or basement installation. Unit has drilled raised-port, cast iron burners and a one-piece 12-gage steel combustion chamber and heat exchanger. Unit has a built-in draft diverter. Easy access is provided to all controls and blower assembly. Standard equipment includes automatic pilot valve, manual main control and manual pilot valve, gas pressure regulator and magnetic type solenoid, with shut-off safety control for L.P. gases. Blower is mounted on tracks for easy removal from the front of the unit and is located below the heat exchanger so that furnace may be installed on floors of combustible materials. Filtered air is provided for summer use. Filter rack is built into the bottom of the unit. A ¼ hp blower motor is used with automatic thermal overload.

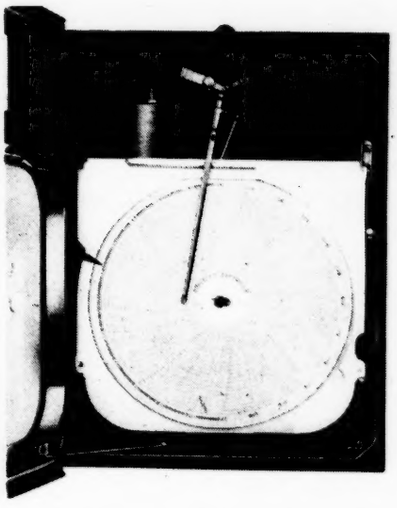


SIZES AND CAPACITIES—Available in two sizes with bonnet output ratings of 75,000 and 87,500 Btu.
MADE BY—*Norge Heat Div., Borg-Warner Corp., Detroit 26, Mich.*

Item 305

Bristol Recording Gage

NAME—Series 500 recording absolute pressure gage.
PURPOSE—Mechanical type recording absolute pressure gage.
FEATURES—Design involves a pressure-sealed lever arm so con-

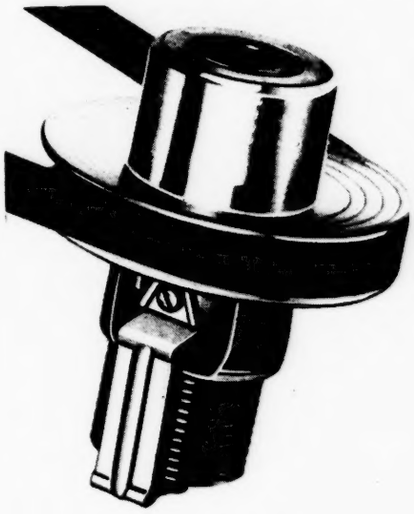


structed that it is free from lost motion and free of turning moments due to changes in differential pressure across the unit. Instrument is said to be rugged in construction and suitable for use under ordinary plant conditions. It provides a record of vacuum in terms of absolute pressure, inherently correct for barometric variations. Instrument is offered as a recorder, as an automatic controller, and for remote measurement and automatic control of absolute pressure.
SIZES AND CAPACITIES — Available in ranges from 0 to 20 millimeters mercury and up.
MADE BY—*The Bristol Co., Waterbury 91, Conn.*

Item 306

Thermodrive Blower Control

NAME—Thermodrive variable speed blower control unit.
PURPOSE—To provide variable speed for blower drives so that speed will

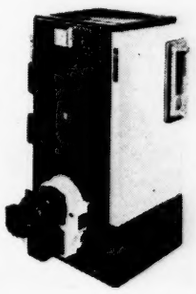


vary with demand on residential forced warm air systems.
FEATURES—A special split-sheave pulley is provided to replace the present blower pulley. Sheaves are held together by spring tension so as to automatically take up belt slack through all stages of the drive's operation. Motor must be rigidly mounted. A standard ½-inch V-belt is used. A 7-foot length of 1½-inch flexible steel tubing is provided to bring the warm air sample from the plenum chamber to the drive control. Tubing runs from the top of the plenum chamber to the unit.
OPERATION—Unit will start the blower at slow speed at the start of the heating cycle and will gradually increase the speed until the maximum is reached. The off-cycle operation is reversed and fan will decelerate and Thermodrive will reach low speed phase just before the fan motor is cut off. At minimum speed the belt will ride on the shaft. At maximum speed it will ride close to the outer edge of the sheaves so that the belt is partially exposed. The two extremes are said to provide an air flow differential of approximately 65% from low to high. As the control air duct temperature increases, the drive will advance the fan speed until the high speed is reached.
SIZES AND CAPACITIES — Available in three models to accommodate standard motor shafts of ½, ⅝ and ¾ inch.
LITERATURE AVAILABLE—Bulletin.
MADE BY—*Webster Electric Co., Racine, Wis.*

Item 307

Smith Boiler-Burner

NAME—Smith Mills 100 boiler-burner.
PURPOSE—Space heating.
FEATURES—A brick chamber has been designed to provide proper reflectivity to the flame developed by the oil burner. Chamber is backed with insulating material so that while temperatures in the heart of the chamber may reach 2,000F, the outside foundation walls remain cool to the touch. Unit is provided with either a tank or tankless heater completely enclosed in the rear section of the boiler for supplying domestic hot water. Boiler sections are made of grey iron. Boiler flue passages are modeled after the Mills boilers so that the extra flue gas travel will provide for greater efficiency.



SIZES AND CAPACITIES — Available in four sizes with I-B-R ratings of from 84,000 to 208,000 Btu per hr and with burner capacity fuel consumption ratings from 1.3 to 2.05 gph. Available for hot water or steam heating systems.
LITERATURE AVAILABLE—Folder.
MADE BY—*The H. B. Smith Co., Inc., Westfield, Mass.*

Item 308

Flowline Welding Fittings

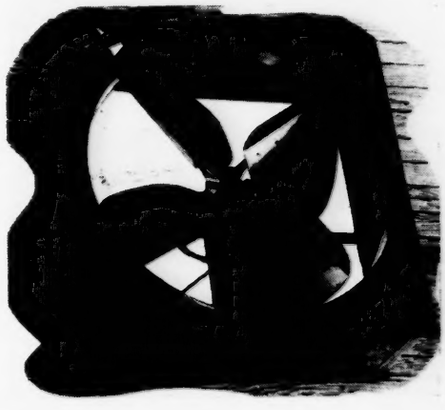
NAME—Flowline stainless steel welding fittings.
PURPOSE—Stainless steel welding fittings in five different ranges of wall thicknesses and sizes, and in three different stainless alloys.
FEATURES—Wall thicknesses of fittings are made to conform to Schedules 5, 10, 40 and 80 specifications in nominal pipe sizes from ½ to 12 inches, O.D. Tube size fittings are made in outside diameters from 1- through 6-inch. Standard stainless alloys are types 304, 316, and 347. Standard fittings include 90 and 45 deg elbows, 100 deg returns, lap joint stub ends, caps, straight tees, and concentric reducers. Other fittings available on special order.
LITERATURE AVAILABLE — Bulletin S-310.
MADE BY—*Welding Fittings Corp., New Castle, Pa.*

Item 309

Emerson Exhaust Fan

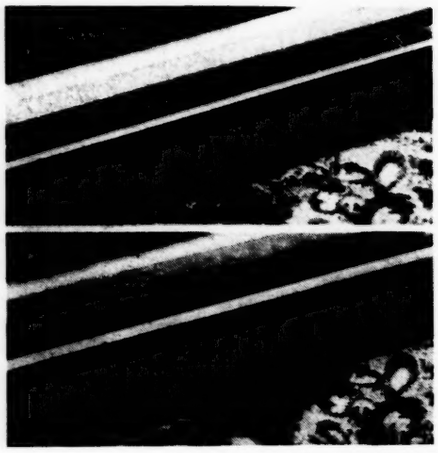
NAME—Emerson exhaust fan.
PURPOSE—Belt-driven fan for horizontal or vertical discharge.
FEATURES—Fan is provided with ball-bearing motors and ball-bearing pillow blocks to absorb the thrust load of the fan so that unit is suitable for installation to discharge vertically, horizontally, or at any angle without injury to the assembly. Fan is powered by a capacitor motor that has resilient rubber hub mounting to insulate noises. Automatic thermal protectors are used to prevent injury to motors due to overloading. Four specially designed blades are shaped, balanced and pitched for large volume air delivery and for quiet operation.
SIZES AND CAPACITIES—36-, 42- and 48-inch sizes. Capacities from 10,200 to 19,350 cfm.
LITERATURE AVAILABLE—Catalog.
MADE BY—*The Emerson Electric Mfg. Co., St. Louis 21, Mo.*

Item 310



Rittling Baseboard Heater

NAME—Rittling baseboard heating unit.
PURPOSE—Space heating.
FEATURES—A new type of deflector is used with front opening to har-

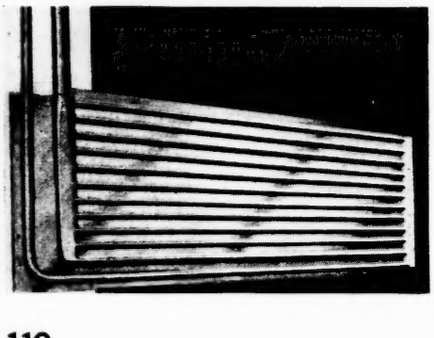


monize with room interiors. Covers and deflectors are fabricated in 6-inch increments to eliminate on-the-job cutting or sawing and to reduce installation time and costs. Extra wide joining pieces permit variations in fitting cover and deflector to proper wall sizes. Brackets are designed to prevent the core from contacting the cover or deflector and thus to eliminate noises due to expansion and contraction. Copper tube with aluminum fins is used.
SIZES AND CAPACITIES—600 Btu per hr per ft with 170F forced water.
LITERATURE AVAILABLE—Catalog.
MADE BY—*The Rittling Corp., 1292 Niagara St., Buffalo 13, N. Y.*

Item 311

Platecoil Heater

NAME—Platecoil.
PURPOSE—For heating or cooling solutions in a tank.
FEATURES — A lightweight, em-

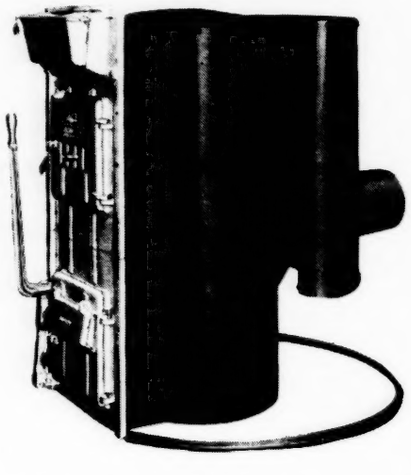


bossed plate is used to carry the heating solution. Device occupies a small space in the tank and is said to have a high Btu capacity per sq ft of area. Coil can be easily cleaned of process coating and sludge. Each plate is tested under water at a pressure of 180 lb per sq in. Unit can also be used for cooling high temperature solutions.
LITERATURE AVAILABLE—Bulletin.
MADE BY—*Kold-Hold Mfg. Co., Lansing 4, Mich.*

Item 312

Apex Furnace

NAME—Apex steel furnace.
PURPOSE—A coal-burning, gravity-type round-cased, hot-air steel furnace for space heating.
FEATURES—The drum or shell is both welded for tightness and riv-

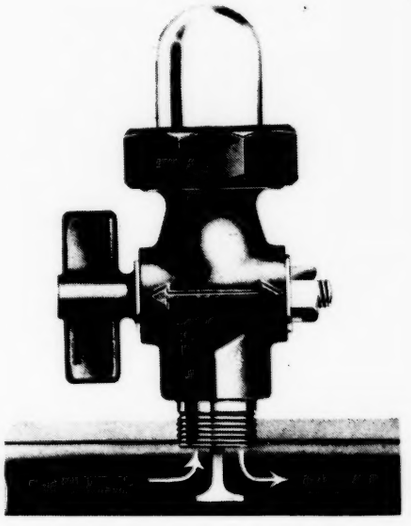


eted for strength so that for the life of the furnace there is no gas or dust leakage. The large surface heat radiator at the rear is also welded and riveted with the smoke pipe outlet located near the bottom. This is said to retard the passage of hot flue gases for greater heating efficiency. Firepot is large and is protected by fire-brick. A large water pan in the top section of the furnace located at the front and extending over the shell and radiator provides humidity. Grate is of the dump-center type, manually operated by a waist-high shaking lever.
SIZES AND CAPACITIES — Available in four sizes from 62,300 to 107,500 Btu at the register. Casing diameter from 40 to 48 inches and overall height, 66 inches.
MADE BY—*The Excelsior Steel Furnace Co., 118 S. Clinton St., Chicago 6, Ill.*

Item 313

Worthington Sight Glass

NAME — Worthington valve type sight glass.
PURPOSE—To provide visibility of



transparent liquids flowing through pipe lines under pressure.
FEATURES—Differential pressure is established across the deflector to allow some of the liquid to circulate continuously through a glass observation bowl. This bowl may be removed for cleaning. Valve is constructed of brass and the observation bowl is made of gage glass material. Unit is designed for pressures up to 50 lb per sq in and for temperatures up to 300F.
SIZES AND CAPACITIES — Overall length, 7 3/16 inches, threaded for ¾-inch standard pipe tap.
MADE BY—*Worthington Pump and Machinery Corp., Harrison, N. J.*

Item 314

Skidmore Pump

In a description of the new type CV Skidmore vertical condensation pump in last month's issue (page 92) the address of the manufacturer was incorrectly given as St. Joseph, Mo. This should have been *Skidmore Corp., St. Joseph, Mich.*

Humidity and Temperature Indicator

NAME — Buensod-Stacey portable electric humidity and temperature indicator.

PURPOSE—To provide direct read-

ings of percentage of relative humidity and temperature in degrees Fahrenheit.

FEATURES—Sensing elements are mounted in a screened metal case and are fastened to an extension rod for taking direct readings at air duct supply grilles or at any other point in an air conditioning system. Accuracy of relative humidity readings, plus or minus 1½%; accuracy of temperature readings, plus or minus ¾F. Response to relative humidity changes in from 10 to 20 seconds at 10 ft velocity. Reading dial has two scales. Upper one is calibrated in per cent relative humidity with a range of 30 to 80% and the lower scale has a range of 40 to 100F.

LITERATURE AVAILABLE—Bulletin.

MADE BY—*Buensod-Stacey, Inc., 60 E. 42nd St., New York 17, N. Y.*

Item 315

Fireball Burner

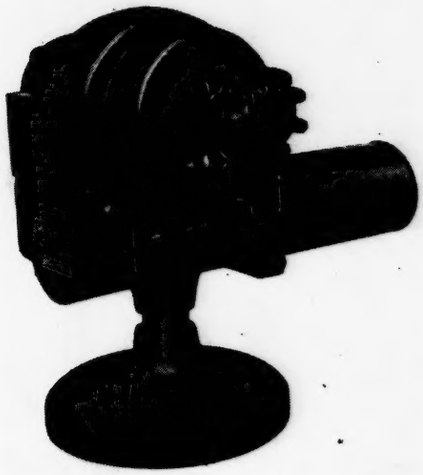
NAME—Fireball oil burner, series A.

PURPOSE—Space heating using oil as fuel.

FEATURES—All parts in straight air tube are accessible from rear of burner. Burner is provided with a Sundstrand fuel pump, Sola transformer, Marathon motor, and Wayne controls. Housing, tube and pedestal are all cast construction. Burner is available in flange and pedestal mountings for original equipment and conversion applications.

SIZES AND CAPACITIES—Oil consumption from ¾ to 1½ gph, and a Btu per hr input of 105,000 to 210,000.

LITERATURE AVAILABLE — Catalog No. 108.



MADE BY—*Wayne Home Equipment Co., Inc., 800 Glasgow Ave., Ft. Wayne 4, Ind.*

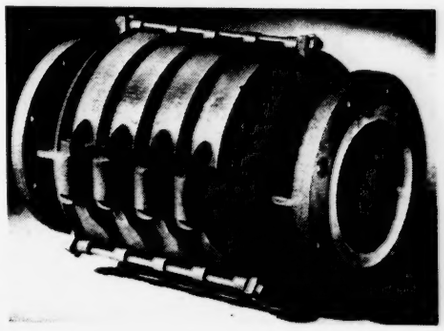
Item 316

Adsco Expansion Joint

NAME — Adsco Corruflex packless expansion joint.

PURPOSE—Packless joint to absorb expansion and contraction due to temperature changes in steam, liquid and gas pipe lines.

FEATURES—Unit requires no packing and makers claim that this results in more economical installa-



tions, since the compactness of design permits use in cramped quarters. Joint is available with single or multiple corrugation, with or without self-equalizing rings, single or double units, and with either flanged or welding ends. It is supplied in copper, stainless steel or other alloys and, where required, with internal sleeves. Traverse of joint ranges from fractions of an inch to 15 inches. Makers claim it will operate under pressures from vacuum to 300 lb and temperatures from sub-zero to 1600F.

SIZES AND CAPACITIES — In sizes from 3 to 24 inches.

LITERATURE AVAILABLE—Folder No. R-149.

MADE BY—*American District Steam Co., North Tonawanda, N. Y.*

Item 317

AUGUST, 1949

TO OBTAIN FURTHER INFORMATION ON NEW EQUIPMENT

On the list below, circle the item number of the equipment in which you are interested, using the number found at the end of each item.

287	288	289	290	291	292	293	294
295	296	297	298	299	300	301	302
303	304	305	306	307	308	309	310
311	312	313	314	315	316	317	

Print your name and address, detach and mail to
Editor, HEATING AND VENTILATING, 148 Lafayette St., New York 13, N. Y.

Name.....

Title (Must be shown).....

Firm (Must be shown).....
(For prompt service, title and firm name must be shown above.)

Business Address.....

City.....**Zone**....**State**..... S 49

NEWS OF THE MONTH

AIR CONDITIONING BUSINESS

approaches quarter billion mark in 1948 but is down slightly from 1947 high except for self-contained units.

Shipments of complete air conditioning equipment and components and accessories for air conditioning and commercial refrigeration equipment during 1948 were valued at $216 million according to the Bureau of the Census, Department of Commerce. This represents a decrease of 8% from the $234 million of this equipment shipped during 1947. Components and accessories for air conditioning and commercial refrigeration equipment amounted to $163 million, or 75%, of the total value of 1948 shipments; complete air conditioning equipment amounted to $48 million or 22%; and ice-making machinery, the remaining $5 million.

• **COMPONENTS AND ACCESSORIES.**—The 1948 value of shipments ($163 million) decreased 15% from the $191 million in 1947, a reversal of the upward movement begun in 1945. The decrease in the value of shipments from 1947 to 1948 is due primarily to the decrease in the value of condensing units and is reflected in each type of condensing unit for which data were collected.

The value of all types of heat exchanger equipment shipped by establishments reporting in the 1948 survey decreased 6%, from $50 million in 1947 to $47 million in 1948. This decrease was reflected in all major types of heat exchanger equipment, except for air conditioning unit coolers.

• **SELF-CONTAINED UNITS.**—Shipments of self-contained air conditioning units continued to increase in 1948 as compared with previous years. Room type air conditioning units increased 56% from $9.9 million in 1947 to $15.5 million in 1948; however, other than room type units increased only 9% from $29.6 million to $32.3 million.

• **EXPORT.**—Factory shipments for export of components and accessories for air conditioning and commercial refrigeration equipment amounted to $12.5 million, or 8% of the total shipments of these products. Exports of complete air conditioning units amounted to $2.2 million or 5% of the total and exports of ice-making machines amounted to $0.3 million or again 5% of the total. The export statistics included in this release represent data reported at the manufacturers' level, based on their knowledge of the ultimate destination of their products and may, therefore, differ from actual exports.

• **GENERAL.**—The statistics included in this report apply to equipment actually billed and shipped. These figures are equivalent to completed sales. Complete units delivered on consignment or shipped to a branch warehouse for stock are not included until such time as they are actually sold. The dollar values reported are the manufacturers' net billing prices, f.o.b. factory.

MID-YEAR REPORT

shows building and construction volume, off 6%, still nears record total. Public building is up 17% to balance drop in private contracts.

F. W. Dodge Corp. has reported that the dollar volume of contracts awarded for building and engineering works in the thirty-seven states east of the Rocky Mountains amounted to $4,467,676,000 in the first half of this year to show a decline of 6% from the corresponding period of last year. The total was the highest first-half volume on record, excepting only last year, the Dodge statistics show.

High lights of the corporation's analysis of building trends in the six-month period follow:

Contracts awarded by government agencies and public bodies showed a 17% increase over the first half of last year with a total of $1,756,243,000.

Private building and engineering contracts totaled $2,711,433,000, down 17% from the private-work total for the first half of last year.

The over-all first-half decline of 6% reflected a 6% drop in nonresidential contracts to a total of $1,744,180,000. Residential awards declined 11% to total $1,624,799,000 while public works and utilities volume totaled $1,098,697,000, a gain of 1% over the corresponding half of last year.

• **PUBLIC.**—The increasing volume of public-account work was the principal characteristic of the half-year trend, and it affected appreciably all major classifications of building and construction, the Dodge analysis shows.

Public-account nonresidential contracts in the first half amounted to $644,417,000 or 37% of all nonresidential building awards. Public agencies awarded residential contracts amounting to $177,661,000 in the first half, and this volume represented 11% of all residential awards. Public-account heavy engineering project awards accounted for 85% of all such contracts and amounted to $934,165,000.

The Dodge corporation reported that record, or near record, volume for a first-half period was shown in the following classifications of building: educational and science, hospitals and institutions, public buildings such as court houses, religious buildings, and social and recreational building.

• **RESIDENTIAL.**—While residential building was down from last year, a significant 3% increase was shown in single family houses built on owners' orders for their own occupancy, with a record volume of $448,204,000. Apartment house awards were down 2% to a total of $385,216,000.

Volume-wise the sharpest decline was shown in single-family houses built by operative builders for sale or rent, the first-half total of $675,278,000 being off 20% from the total for the corresponding half of last year.

AIR CONDITIONING STATISTICS

show that Houston is controlling its weather, probably has highest cooling tonnage in U. S.

Approximately $75,000,000 has been spent in making Houston one of the top cities—if not the first—in air-conditioning in the nation. Houston now has approximately 107,000 tons of air-conditioning, requiring approximately 120,000 horse power daily.

And these figures do not include attic and window ventilation. When this class of cooling system is included, the total spent on cooling facilities in this city would approximate $100,000,000 and raise the daily horsepower to probably 140,000.

A survey made by Houston Magazine, publication of the Houston Chamber of Commerce, the last week in June revealed that in excess of 77,000 tons of air-conditioning equipment had been installed here since 1946. According to statistics available, no other city has a better record. This survey was made with the cooperation of the local air-conditioning engineers, distributors, dealers and branches of air-conditioning equipment manufacturers. Prior to 1946, according to the files of the magazine, the city processed approximately 30,000 tons of air-conditioning.

• **HOMES.**—As John E. Haines, vice-president of the Minneapolis-Honeywell Regulator Co., pointed out on a recent visit, Houston not only has a record amount of air-conditioning tonnage, but its mechanical cooling reaches into every section of the city and into not only most business and public places, but into its homes.

More than 7500 Houston homes are partially air-conditioned, more than 1000 are completely air-conditioned, and approximately 1200 have systems of five tons or more. Many small, fraction ton units of which no available record has been revealed would probably considerably raise the number of homes in the city which have one or more rooms air-conditioned. In addition to the number mentioned, many thousands of homes are cooled by attic and window fan ventilation.

Residential air conditioning accounted for the largest number of jobs (1990) and largest total tonnage (5425) installed in 1949 of any classification covered by the survey, with office buildings second at 4823 tons.

• **OFFICES.**—Approximately forty-five office buildings in the city are completely air-conditioned, as are more than 500 individual offices and office suites outside of the forty-five air-conditioned buildings. In fact, most of the business offices of downtown Houston are either air-conditioned or in the process of being air-conditioned. One large office building recently spent nearly as much for remodeling and air-conditioning as the building originally cost a decade ago. If the present trend continues, within the very near future, a warm office will be an oddity in the city.

One hundred and thirty-seven restaurants, cafes or eating places are well air-conditioned and many others are partially cooled. More than three dozen churches and a dozen or more clubs are cooled for comfort, as are twenty-seven department stores, more than a half hundred drug stores, several hundred doctors' and dentists' offices, a dozen banks and hundreds of retail stores of many classifications.

• **HOTEL ROOMS.**—Houston now has 3435 air-conditioned hotel rooms and the final 150 of the 500 rooms at the Lamar Hotel will be completed in September, making a total of 3585.

Many of the motor hotels of this city are also air-conditioned. The courts have a total of 286 air-conditioned rooms and several are now adding more air-cooled rooms. Many courts also have attic or window ventilation.

• **STORES.**—The majority of this city's retail stores are now air-conditioned. So are the cafes, the places of amusement and the doctors', dentists' and business offices.

A summary of the magazine Houston's recent survey is given in the table:

	Jobs	Tons
1946	366	7,959
1947	1,063	15,526.5
1948	2,996	25,925.25
1949 (to June 30)	2,382	27,829.5
	6,807	77,240.25

WARM AIR HEATING

sales and research emphasized at mid-year convention of National Warm Air Heating and Air Conditioning Association.

Nearly 400 warm air heating men representing 175 companies attended the mid-year convention of the National Warm Air Heating and Air Conditioning Association held at the Edgewater Beach Hotel, Chicago, June 21 and 22.

The board of trustees nominated and elected to the board H. F. Randolph, vice president, International Heater Co., to fill the unexpired term of E. H. Paul, resigned.

• **RESEARCH.**—Following the call to order on June 21 by the association's president, Atlee Wise, Wise Furnace Co., Akron, Ohio, Robert W. Roose, special research assistant in mechanical engineering at the University of Illinois, discussed the results of research activities conducted in Research Residence No. 2 during the heating season of 1948-49. The title of his talk was Ceiling Panel Heating versus Conventional Forced Warm Air Heating with an Unheated Basement. Mr. Roose compared the performance with previous tests of the same system conducted with a heated basement. From the tests conducted, Mr. Roose concluded that the two heating systems performed satisfactorily and had approximately the same characteristics with either a heated or unheated basement. In addition, the extra fuel required to heat the basement was relatively small compared to the additional space heated.

Norman A. Buckley, also of the University of Illinois, described recent research activities conducted in the mechanical engineering laboratories. Mr. Buckley explained the proposed arrangement and method for determining the performance of a blower used with a furnace.

• **SALES.**—In his talk on marketing opportunities, Larry S. Redford, vice president of Jackson & Church, pointed out that although total square footage of residential heating in 1949 is below 1948 level, the square footage of nonresidential structures will be above the 1948 level. He mentioned that hospitals and institution buildings are up 3%, religious buildings are up 19%, dormitory buildings are up 57% and public buildings of all types are up 96%.

• **NEW RESIDENCE.**—F. L. Meyer, president of the Meyer Furnace Co. and chairman of the Research Advisory Committee, announced the construction of the new Research Residence No. 3, being built by the Association at Champaign, Illinois. Morris E. Childs of the University of Illinois described the home layout and heating apparatus to be used in the new residence. This home will be a typical 5-room basementless structure now selling with lot from $9,000.00 to $10,000.00 of standard frame construction. It will have a wood shingle exterior. Over-all dimensions of the home are 32 ft x 24 ft providing 778 sq ft of floor area. Floor construction will consist of a 4-inch gravel fill and a 4-inch concrete slab with a vapor barrier between the gravel and the slab. The slab will be insulated with 1-inch rigid insulation between the slab and the foundation.

Plans provide for the testing of three types of systems during the 1949-50 heating season. One is a gravity warm air system with high side wall registers in each room and two return air grilles in the utility room walls. The second heating system will be a forced warm air system using the same duct work as with the gravity system and the third test will be conducted with a forced warm air system with low wall registers in each room and three side wall return air grilles. These three systems will be operated alternately over two-week periods throughout the season in order that comparative performance may be studied under all types of weather conditions.

NEWS BRIEFS

• The number of mechanical stokers sold in March 1949 was approximately one-third of the number sold in March 1948, and one-half the number sold in 1947. Of the 1,897 units sold in March, 1,456 were sold for residential use and 156 for small apartment houses and commercial heating plants.

• The Engineering Committee of the Industrial Unit Heater Association is carrying on a series of laboratory tests in the preparation of a uniform Code for Sound Measurement of Unit Heaters.

• A 10% to 15% drop in building costs in metropolitan New York from the postwar peak of September, 1948 was reported by the New York Building Congress.

• Tests of properties of lightweight aggregate concretes by the National Bureau of Standards showed coefficients of thermal conductivity (k values) ranging from 1.7 to 2.4 Btu (inch) per (hr) (sq ft) (F) for the pumice concretes as compared with values of 6.7 and 9.6 for sand and gravel concrete. Still lower k values (0.75 to 1.6) were measured on vermiculite and perlite concretes. These latter two materials were lightest tested but show low strength and high porosity.

• A Missouri pig breeder, W. A. Heydenberk of Barton County, keeps the pressure water system across his barnyard from freezing by soil heating cable in an 18-foot long trough. The cable is placed in the bottom of the trough and protected by a built-in covering. A thermostat turns the heat on when the water temperature approaches the freezing point.

• The Anthracite Standards Law of Pennsylvania has been amended by the 1949 session of the Legislature so that, effective September 1, it will become mandatory for anthracite producers and Pennsylvania dealers to lable all sizes of anthracite as "standard anthracite" or "sub-standard anthracite," as the case may be.

• The American Public Health Association in cooperation with the National Security Resources Board, is preparing a roster of sanitary and public health engineer citizens of the United States. Basic definition of a sanitary engineer is that prepared by the National Research Council in 1943. Each individual will decide his own qualifications within that description. Questionnaires will be circulated among known sanitary engineers, and those who do not receive a questionnaire within the next two or three months are requested to inform the Engineering Section Project, American Public Health Association, 1790 Broadway, N. Y.

• Results of a single test installation led to the air conditioning of a chain of super markets. Typhoon Air Conditioning Co., Inc. of Brooklyn, N. Y., reports that Shopwell Foods, Inc., which placed Typhoon units in its Scarsdale outlet in 1948, have now ordered units for a total of 10 stores, with additional installation in the planning stage.

• Officers elected for the year 1949 to 1950 at the Hot Springs, Va., meeting of the Air Conditioning & Refrigerating Machinery Association were: W. H. Aubrey, Frick Co., Inc., president; A. P. Shanklin, Carrier Corp., first vice president; Harold F. Smiddy, second vice president; G. A. Heuser, Henry Vogt Machine Co., treasurer; George S. Jones, Jr., Servel Inc., chairman of the executive committee.

DEGREE-DAYS FOR JUNE, 1949

(A) Airport readings; (C) City office readings; (O) Readings at a point on outskirts of city.

HEATING AND VENTILATING'S 21st Year of Publication of Monthly Degree-Day Data

City	June			Cumulative, Sept. 1 to June 30			Heating Season[1]
	1949	1948	Normal	1948-49	1947-48	Normal	Normal
Abilene, Texas (A)	0	0	0	2818	2853	2061	2061
Albany, New York (A)	34	75	0	6162	7473	6580	6580
Albuquerque, New Mexico (A)	3	3	0	4640	4604	4298	4298
Alpena, Michigan (C)	100	127	138	7124	8011	8265	8299*
Anaconda, Montana (C)	281	230	251	8897	8489	8261	8357**
Asheville, North Carolina (C)	3	5	0	3516	3908	4232	4232
Atlanta, Georgia (A)	0	0	0	2365	2830	2890	2890
Atlantic City, New Jersey (C)	18	27	0	4168	5092	5176	5176
Augusta, Georgia (C)	0	0	0	1620	2289	2161	2161
Baker, Oregon (C)	205	121	213	7800	6855	7163	7163
Baltimore, Maryland (C)	0	0	0	3670	4469	4533	4533
Billings, Montana (A)	106	96	60	7591	7090	7119	7119
Binghamton, New York (C)	37	54	0	5831	6754	6808	6808
Birmingham, Alabama (A)	0	0	0	2285	2765	2352	2352
Bismarck, North Dakota (A)	77	96	45	9125	9341	9192	9192
Block Island, Rhode Island (C)	0	136	99	4940	6091	5788	5788
Boise, Idaho (A)	72	44	0	6540	6010	5552	5552
Boston, Massachusetts (A)	20	113	0	4999	6275	6045	6045
Bozeman, Montana (C)	253	181	230	8594	8218	8425	8521**
Buffalo, New York (A)	39	71	12	5925	6768	6822	6822
Burlington, Iowa (A)	1	10	(a)	6050	6182	(a)	(a)
Burlington, Vermont (A)	44	114	3	6845	8184	7514	7514
Butte, Montana (C)	266	189	236	9407	8881	8162	8235**
Cairo, Illinois (C)	0	0	0	3441	3945	3909	3909
Canton, New York (C)	55	122	0	7068	8270	8020	8020
Charleston, South Carolina (C)	0	0	0	1166	1944	1769	1769
Charlotte, North Carolina (C)	0	0	0	2451	3177	3120	3120
Chattanooga, Tennessee (A)	0	0	0	2848	3239	3118	3118
Cheyenne, Wyoming (A)	176	154	126	7840	7519	7466	7466
Chicago, Illinois (C)	23	61	59	5579	6154	6077	6077
Cincinnati; Ohio (C)	1	1	0	3992	4571	4684	4684
Cleveland, Ohio (A)	26	40	0	5435	6090	6155	6155
Columbia, Missouri (C)	0	0	0	4772	4830	4922	4922
Columbia, South Carolina (C)	0	0	0	1767	2495	2364	2364
Columbus, Ohio (C)	7	3	0	4691	5291	5398	5398
Concord, New Hampshire (A)	52	157	54	6608	8031	7353	7353
Concordia, Kansas (C)	0	3	0	5759	5516	5315	5315
Dallas, Texas (A)	0	0	0	2473	2693	2256	2256
Davenport, Iowa (C)	6	7	0	5899	6172	6289	6289
Dayton, Ohio (A)	5	12	0	5192	5799	5264	5264
Deer Lodge, Montana (C)	235	190	235	9141	8196	8582	8672**
Denver, Colorado (C)	72	57	0	6009	5828	5874	5874
Des Moines, Iowa (C)	3	6	0	6311	6288	6384	6384
Detroit, Michigan (A)	31	52	0	5828	6554	6490	6490
Devils Lake, North Dakota (C)	122	112	96	9992	10,257	9970	9970
Dodge City, Kansas (A)	6	7	0	5489	5402	5035	5035
Dubuque, Iowa (C)	9	29	0	6467	6706	6790	6790
Duluth, Minnesota (C)	128	191	216	8698	9503	9483	9483
Eastport, Maine (C)	188	341	300	7227	8373	8241	8520**
Elkins, West Virginia (A)	40	49	0	5296	5633	5697	5697
El Paso, Texas (A)	0	0	0	2930	2908	2428	2428
Ely, Nevada (A)	(a)	227	(a)	(a)	7971	(a)	(a)
Erie, Pennsylvania (C)	30	54	0	5395	6120	6273	6273
Escanaba, Michigan (C)	89	161	138	7758	8558	8746	8771
Evansville, Indiana (A)	0	0	0	4208	4739	4244	4244
Fort Smith, Arkansas (A)	0	0	0	3250	3437	3147	3147
Fort Wayne, Indiana (A)	19	20	0	5710	6388	5925	5925
Fort Worth, Texas (A)	0	0	0	2438	2720	2148	2148
Fresno, California (A)	0	0	0	2755	2611	2334	2334
Galveston, Texas (C)	0	0	0	993	1445	1016	1016
Grand Junction, Colorado (A)	41	30	0	6341	6019	5548	5548
Grand Rapids, Michigan (C)	23	45	0	5891	6512	6535	6535
Green Bay, Wisconsin (C)	37	79	0	7343	8003	7825	7825
Greensboro, North Carolina (A)	1	2	0	3260	3841	3529	3529
Greenville, South Carolina (A)	0	0	0	2618	3166	3380	3380
Harrisburg, Pennsylvania (A)	10	10	0	4834	5500	5375	5375
Hartford, Connecticut (A)	20	57	0	5413	6635	6036	6036
Hatteras, North Carolina (C)	0	1	0	1891	2422	2571	2571
Havre, Montana (C)	82	112	144	8653	7910	8700	8700
Helena, Montana (A)	183	153	171	9013	8015	7834	7894**
Houston, Texas (C)	0	0	0	1283	1557	1157	1157
Huron, South Dakota (A)	24	61	0	7975	8228	8004	8004
Indianapolis, Indiana (A)	4	4	0	5148	5310	5298	5298
Jackson, Mississippi (A)	0	0	(a)	1805	2296	(a)	(a)
Kansas City, Missouri (A)	0	0	0	4874	4962	4956	4956
Knoxville, Tennessee (A)	0	0	0	3126	3442	3670	3670
La Crosse, Wisconsin (A)	15	38	0	7392	7866	7322	7322
Lander, Wyoming (A)	168	106	135	8513	7783	7947	7947

(a) Data not available.

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

Figures in this table, with seven exceptions, based on local weather bureau reports. Exceptions are Utica and Lewiston, figures for which are furnished through the courtesy of Coke Sales Department, Central New York Power Corp., Utica, N. Y., and Norman E. Ross, Bursar, Bates College, Lewiston Me., respectively; Anaconda, Bozeman, Butte, Deer Lodge and Livingston, Mont., through the courtesy of the Montana Power Company.

[Table concluded on page 116]

Degree-Days for June, 1949 (Concluded)

(A) Airport readings; (C) City office readings; (O) Readings at a point on outskirts of city.

HEATING AND VENTILATING's 21st Year of Publication of Monthly Degree-Day Data

City	June			Cumulative, Sept. 1 to June 30			Heating Season[1]
	1949	1948	Normal	1948-49	1947-48	Normal	Normal
Lansing, Michigan (A)	31	84	0	6417	7096	7048	7048
Lewiston, Maine (O)	(a)	138	45	(a)	7848	7707	7707
Lincoln, Nebraska (C)	4	2	0	6291	6019	5999	5999
Little Rock, Arkansas (A)	0	0	0	2847	3223	2811	2811
Livingston, Montana (C)	179	149	142	7566	7489	7185	7245**
Los Angeles, California (C)	0	17	0	1860	1548	1504	1504
Louisville, Kentucky (A)	0	0	0	3938	4351	4180	4180
Lynchburg, Virginia (A)	2	3	0	3675	4262	3980	3980
Macon, Georgia (C)	0	0	0	1770	2337	2201	2201
Madison, Wisconsin (C)	27	51	0	6929	7483	7429	7429
Marquette, Michigan (C)	106	220	186	7775	8512	8650	8693*
Memphis, Tennessee (A)	0	0	0	2818	3371	2950	2950
Meridian, Mississippi (C)	0	0	0	1822	2336	2160	2160
Milwaukee, Wisconsin (A)	41	88	39	6688	7241	7245	7245
Minneapolis, Minnesota (A)	17	24	0	7624	8032	7850	7850
Montgomery, Alabama (C)	0	0	0	1509	2017	1884	1884
Nantucket, Massachusetts (A)	75	212	123	5384	6397	5957	5957
Nashville, Tennessee (A)	0	0	0	3229	3664	3507	3507
New Haven, Connecticut (A)	34	64	0	5189	6402	5895	5895
New Orleans, Louisiana (C)	0	0	0	900	1402	1024	1024
New York, New York (C)	9	16	30	4250	5373	5274	5274***
Nome, Alaska† (A)	1108	1013	973	13,604	13,095	13,058	14,580**
Norfolk, Virginia (C)	0	6	0	2650	3338	3350	3350
North Head, Washington (C)	343	226	309	5352	4865	5002	5452**
North Platte, Nebraska (C)	31	36	0	6804	6169	6366	6366
Oakland, California (A)	87	120	93	3370	3276	2963	3143**
Oklahoma City, Oklahoma (C)	0	0	0	3693	3821	3613	3613
Omaha, Nebraska (A)	2	3	0	6436	6220	6131	6131
Oswego, New York (C)	55	129	63	6118	7105	7088	7088
Parkersburg, W. Virginia (C)	6	8	0	4291	4756	4775	4775
Peoria, Illinois (A)	37	0	0	5765	6012	6109	6109
Philadelphia, Pennsylvania (C)	1	3	17	3969	4916	4737	4737***
Phoenix, Arizona (C)	0	0	0	1818	1651	1405	1405
Pittsburgh, Pennsylvania (C)	11	11	0	4515	5461	5235	5235
Pocatello, Idaho (A)	106	73	60	7476	7007	6655	6655
Portland, Maine (A)	86	184	81	6740	7856	7218	7218
Portland, Oregon (C)	82	49	90	4606	4191	4469	4469
Providence, Rhode Island (C)	17	65	0	4805	5966	6015	6015
Pueblo, Colorado (A)	40	22	0	5846	6047	5514	5514
Raleigh, North Carolina (C)	0	0	0	2606	3363	3234	3234
Rapid City, South Dakota (A)	92	114	30	7685	7215	7118	7118
Reading, Pennsylvania (C)	6	8	0	4456	5296	5389	5389
Red Bluff, California (A)	0	14	(a)	3019	3093	(a)	(a)
Reno, Nevada (A)	114	127	90	6741	6328	5892	5892
Richmond, Virginia (C)	1	1	0	3149	3920	3695	3695
Rochester, New York (A)	49	75	0	6110	6933	6732	6732
Roseburg, Oregon (C)	79	55	114	4565	4410	4428	4428
Roswell, New Mexico (A)	0	0	0	3858	3951	3484	3484
Sacramento, California (C)	5	17	0	2981	2993	2653	2653
St. Joseph, Missouri (A)	0	2	0	5483	5471	5161	5161
St. Louis, Missouri (C)	0	0	0	4311	4622	4585	4585
Salt Lake City, Utah (A)	54	43	0	6713	6022	5555	5555
San Antonio, Texas (A)	0	0	0	1626	1859	1202	1202
San Diego, California (A)	1	26	6	1901	1637	1645	1645
Sandusky, Ohio (C)	20	28	0	5293	6000	6208	6208
San Francisco, California (C)	214	175	195	3269	2922	2876	3264**
Sault Ste. Marie, Michigan (A)	147	223	189	8352	9111	9139	9285**
Savannah, Georgia (A)	0	0	0	1099	1729	1490	1490
Scranton, Pennsylvania (C)	21	47	0	5409	6309	6129	6129
Seattle, Washington (C)	138	84	162	4923	4562	4857	4934**
Sheridan, Wyoming (A)	126	101	120	7987	7508	8008	8008
Shreveport, Louisiana (A)	0	0	0	2029	2454	1938	1938
Sioux City, Iowa (A)	3	22	0	7237	6927	6898	6898
Spokane, Washington (A)	132	66	81	7541	6809	6355	6355
Springfield, Illinois (C)	0	0	0	4926	5162	5373	5373
Springfield, Missouri (A)	0	0	0	4488	4684	4428	4428
Syracuse, New York (A)	37	66	12	6033	7108	6893	6893
Tacoma, Washington (C)	174	104	183	5294	4951	5066	5181**
Terre Haute, Indiana (A)	2	1	0	4946	5460	4872	4872
Toledo, Ohio (A)	23	34	0	5750	6494	6077	6077
Topeka, Kansas (C)	0	2	0	5013	5060	4969	4969
Trenton, New Jersey (C)	10	8	0	4389	5341	4933	4933
Utica, New York (O)	21	66	0	5666	7285	6796	6796
Valentine, Nebraska (C)	41	61	0	7428	7052	7039	7039
Walla Walla, Washington (C)	51	12	0	5629	4982	4808	4808
Washington, D. C. (C)	0	0	0	3640	4368	4626	4626
Wichita, Kansas (A)	0	1	0	4868	4765	4673	4673
Williston, North Dakota (C)	107	(a)	54	9198	(a)	9323	9323
Winnemucca, Nevada (C)	73	73	117	6961	6394	6391	6427**
Yakima, Washington (A)	78	19	0	6567	5840	5599	5599

(a) Data not available.
†Nome data are for May.

*Includes August.
**Includes July and August.
***New 48-year normal covering 1898 to 1946.

[1]Figures in this column are normal totals for a complete heating season, September to June, incl.

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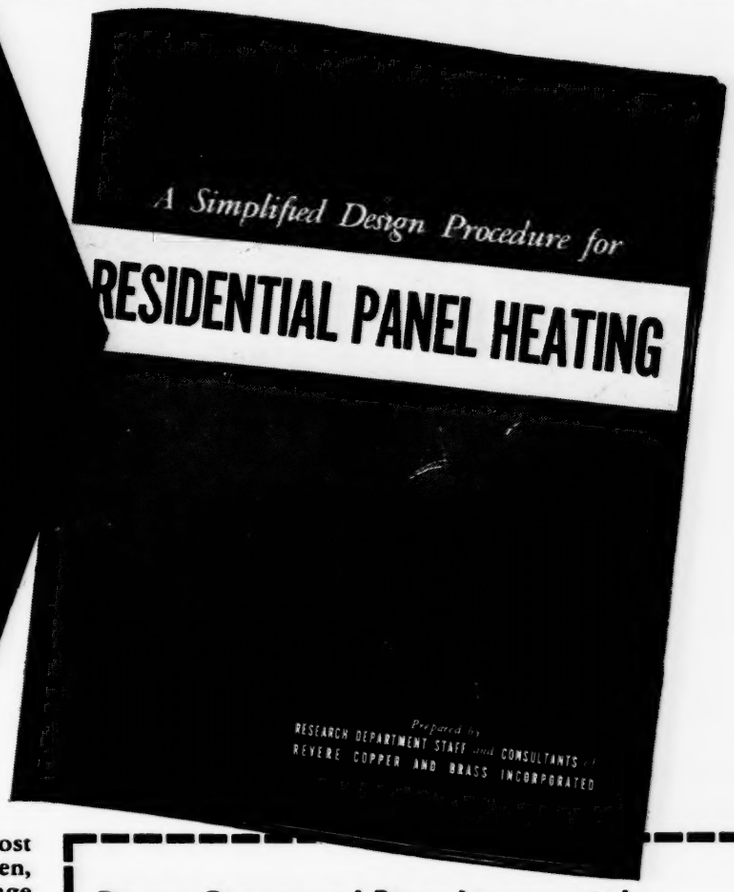
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Abstracts and Reviews

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LAY-OUT BOOK

To assist those in the plumbing and allied pipe trades to compute the approximate measurements for assembling cast iron pipe soil fittings, Percy M. Forster has assembled concise helpful information. Measurements in the book can also be used for copper waste fittings or screw drainage fittings. The volume covers 2-, 3- and 4-inch soil fittings.

On each page is an illustration to show the pipe layout, information on the existing or known measurements, and simple factors to multiply known values in order to obtain desired measurements. Included also are tables to supply measurements for various size cast iron pipe soil fittings, and tables showing measurements for several degrees of bends for the common pipe sizes.

Forster's Practical Layout Book, by Percy M. Forster. Cloth bound, 4¾ x 6¾ inches, 216 pages. Published by Percy M. Forster, P. O. Box 233, Berkeley, Calif. Price, $5.

• • •

COMFORT AIR CONDITIONING—Pertinent information for the man who figures air conditioning loads is contained in the 64-page book developed by the Committee on Air Conditioning of the Heating, Piping and Air Conditioning Contractors National Association. This book on Comfort Air Conditioning is Part IV of the engineering standards issued by the association. Besides containing load calculation procedure, the book contains thermodynamic tables and design information of value to the one concerned with making heat gain estimates. Available to non-members at $2 per copy, it can be purchased through any of the local associations of the Heating, Piping and Air Conditioning Contractors National Association and not through the national association.

COPPER AND BRASS PIPE—Simplified Practice Recommendation R217-49, for Copper Water Tube and Copper and Brass Pipe, has been issued by the Commodity Standards Division of the National Bureau of Standards. Established in 1946, the recommendation consists of a simplified list of types and sizes of copper water tube and copper and brass pipe. Dimensions and weights for 18 sizes of tube and pipe are given as well as dimensional tolerances and recommended uses for the three types of tube covered. Write the Superintendent of Documents, Government Printing Office, Washington 25, D. C. Price, 10 cents each.

BASEBOARD HEATING SYSTEMS—A new I-B-R booklet, Guide No. 5, has been issued by The Institute of Boiler and Radiator Manufacturers, which presents data that will be helpful in the selection and installation of baseboards in one-pipe forced circulation hot water heating systems. This 28-page booklet presents data on the installation of these baseboards with other types of heating systems, such as two-pipe forced circulation or gravity hot water and two-pipe steam. This publication contains a large number of illustrations and tables of value to the heating engineer. The Institute of Boiler and Radiator Manufacturers, 60 E. 42nd St., New York 17, N. Y. Price, 50 cents.

HEALTH ASPECTS OF AIR POLLUTION—A paper on the health aspects of air pollution by Dr. A. J. Lanza, formerly associate medical director of the Metropolitan Life Insurance Company and now chairman of the Institute of Industrial Medicine, New York University-Bellevue Medical Center, was presented at the annual conference, Smoke Prevention Association of America, Birmingham, Alabama. The paper covers the facts relative to air pollution and its medical aspects and contains statistical information regarding cancer deaths from respiratory diseases. For a copy of this paper, write to Smoke Prevention Association of America, Inc., City Hall Square Building, Chicago 2, Ill.

INDUSTRIAL HYGIENE PROBLEMS IN BOLIVIA, PERU AND CHILE—This report by J. J. Bloomfield, Assistant Chief of the Division of Industrial Hygiene, Public Health Service, is based on a thorough investigation of industrial life in the countries listed. The studies were undertaken under the auspices of the Institute of Inter-American Affairs. Various socioeconomic factors as well as industrial hygiene considerations were studied in making this survey and in establishing the recommendations listed. For a copy of this 139-page, paper bound book, write to the Superintendent of Documents, Washington 25, D. C. Price, 40 cents.

PIPES, DUCTS, AND FITTINGS—Simplified Practice Recommendation R207-49 has been issued by the U. S. Department of Commerce on pipes, ducts and fittings for warm air heating and air conditioning. These standards were drawn to establish standard methods of tests, rating, classification and labeling. The recommendation, which covers 24 pages, replaces those that were issued on March 1, 1945. For a copy, write to the Superintendent of Documents, U. S. Government Printing Office, Washington 25, D. C. Price, 10 cents.

FLY ASH—Data on how to reduce the emission of fly ash from small boiler plants is contained in a booklet What Can the Small Plant Do About Fly Ash, that is being distributed by Bituminous Coal Research, Inc. Information is given to provide a basis for judging the amount of cinder the small plant probably is emit-

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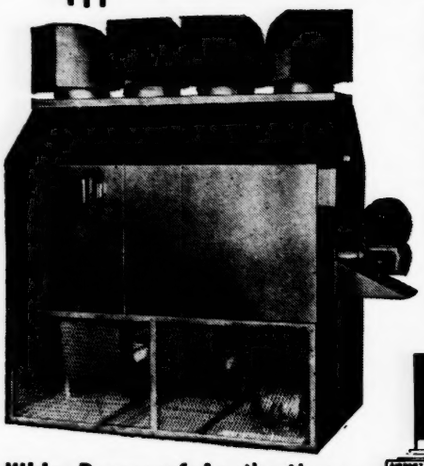

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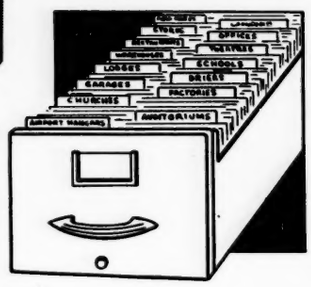

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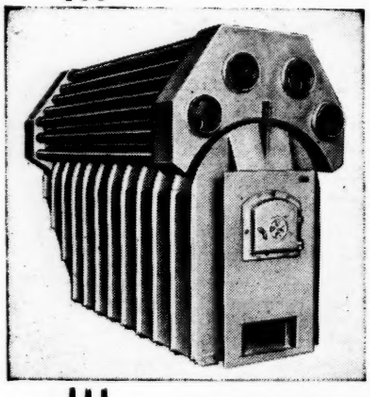
Model Number	Btu	Cfm	hp	Dimensions (inches) length-width-height			Approximate Shipping Wt., Lbs.
T.D.— 25	250,000	3,600	¾	60	32	81	1,300
T.D.— 40	400,000	5,400	1	60	32	81	1,350
T.D.— 50	500,000	6,600	1½	80	32	81	1,780
T.D.— 70	750,000	8,800	2	80	32	81	1,855
T.D.— 80	800,000	10,200	3	80	48	81	2,110
T.D.—100	1,000,000	12,500	5	80	48	81	2,200
T.D.—125	1,250,000	15,300	5	100	54	103	3,000
T.D.—150	1,500,000	19,400	7½	100	54	103	3,250

NATIONAL HEATER COAL FIRED FORCED AIR UNIT



Specifically constructed and designed for stoker and hand firing with blower position at rear of casing, this unit embodies all the features of other NATIONAL heaters for greater satisfaction and economy of operation. Can be quickly and efficiently converted to light oil, heavy oil or gas firing as future fuel costs and supplies may necessitate.

• *Write for Literature*

NATIONAL HEATER CO.
CLEORA & VANDALIA STREETS
ST. PAUL 4, MINNESOTA

ting with different types of firing equipment, both as to dust loading and size of the dust. Graphs enable a ready estimate of the approximate cost to install a cinder collector to provide low stack emission. Bituminous Coal Research, Inc., 912 Oliver Building, Pittsburgh 22, Pa. Price, 25 cents.

TUBERCULOSIS IN INDUSTRY—A panel discussion on tuberculosis in industry was held last year in connection with the 13th annual meeting of the Industrial Hygiene Foundation at Mellon Institute, Pittsburgh. The five panel papers are now published as Bulletin 9 in the Foundation's medical series. It was pointed out that tuberculosis is still a serious problem in large industrial communities. Industrial Hygiene Foundation, 4400 Fifth Avenue, Pittsburgh 13, Pa. Price, 50 cents.

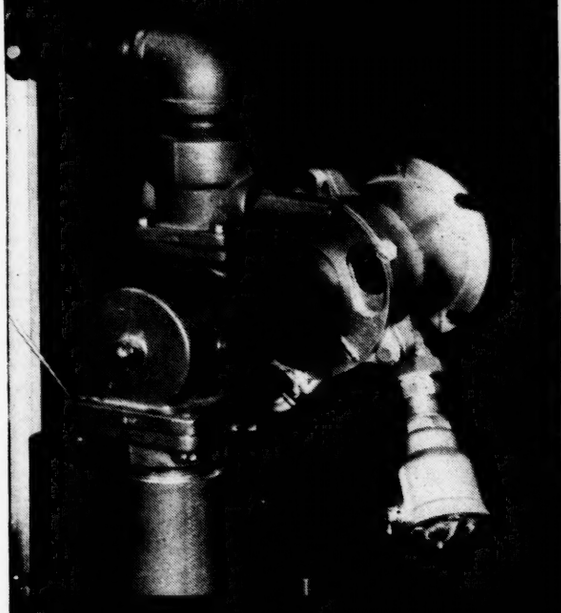
OIL HEATING PRACTICE—The Federal Trade Commission has issued a 9-page booklet on trade practice rules for the oil heating industry of the New England states. The rules are directed to the elimination and prevention of unfair trade practices to the end that the industry, trade, and the public may be protected from harmful effects of such competitive methods. It covers the question of misrepresentation, deceptive advertising, selling below cost, discrimination, and guarantees. Federal Trade Commission, Washington, D. C.

BURNING COAL OR COKE—Based on a study that has been going on at the University of Illinois for the past three years under the direction of Rudard A. Jones, research associate professor of architecture, Circular Series G3.61 has been issued on Homes Planned for Coal or Coke. It covers way to simplify the handling of coal or coke in homes and it explains how to plan in advance for either fuel or ash handling. Included in this 12-page circular are plans for a one-story basementless house, a one-story house with partial basement, a split level house and a two-story house. In each case special features of fuel handling and ash removal are discussed. Small Homes Council, University of Illinois, Urbana-Champaign, Illinois.

UNIFIED AND AMERICAN SCREW THREADS—The new American Standard Unified and American Screw Threads for Screws, Bolts, Nuts, and Other Threaded Parts (ASA Bl. 1-1949) has been published by the American Society of Mechanical Engineers. This standard, approved by the American Standards Association on February 23, 1949, is the American fulfillment of the Declaration of Accord with respect to the Unification of Screw Threads signed at Washington, D. C., November 18, 1948, by British, Canadian and American representatives. The significant thing about the new classes of tolerance is that, unlike the old, which were independently developed from practical experience, the new seek to relate the classes to one another and to take into account the relative influence of the factors of diameter, pitch and length of engagement. Paper bound, 99 pages; price, $3. American Society of Mechanical Engineers, 29 West 39th St., New York 18, N. Y.

MIXJECTOR BLAST

Bryant Industrial TORCH BURNERS



COMPLETE GAS FIRING ASSEMBLIES FOR FURNACES, BOILERS AND KILNS

Mixjector Torches provide two big features: Specially-developed nozzle designed to hold flame tightly at all pressures, plus the Bryant Mixjector, most compact proportional mixer ever designed. Available in 9 sizes, capacities to 6,750,000 Btu per hour with 16-ounce air. *Write for Data Sheet 2C-1 for complete description.*

BRYANT INDUSTRIAL DIVISION
Affiliated Gas Equipment, Inc.
1020 LONDON ROAD • CLEVELAND 10, OHIO

GENERAL G C **CONTROLS**

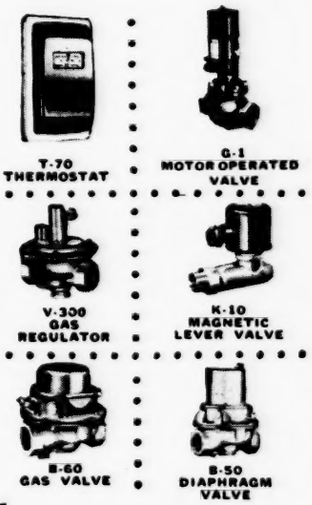
Takes the MANual out of Controls

They're Automatic!

For all *Pressure, Temperature, Level and Flow Control* applications using all kinds of gases, liquids, and air.

It's profitable! Because antiquated automatic or manually operated controls hold down your profits! That's true whether you are a manufacturer, wholesaler, dealer or user.

- "ALL-GAS" CONTROL SYSTEMS
- AIRCRAFT CONTROLS, ELECTRIC
- AUTOMATIC SAFETY SHUT-OFF VALVES
- MOTOR OPERATED VALVES
- MAGNETIC GAS VALVES
- SOLENOID VALVES
- THERMOSTATIC CONTROLS & CONTROL SYSTEMS
- GOVERNOR-TYPE VALVES
- HI-LOW-OFF VALVES
- DIAPHRAGM VALVES
- LIMIT CONTROLS
- ELECTRIC TIME SWITCHES
- REFRIGERATION VALVES
- GAS REGULATORS
- LOW WATER CONTROLS
- STRAINERS, RELAYS
- TRANSFORMERS, SWITCHES
- PORTABLE EQUIPMENT CONTROLS
- MACHINE TOOL CONTROLS

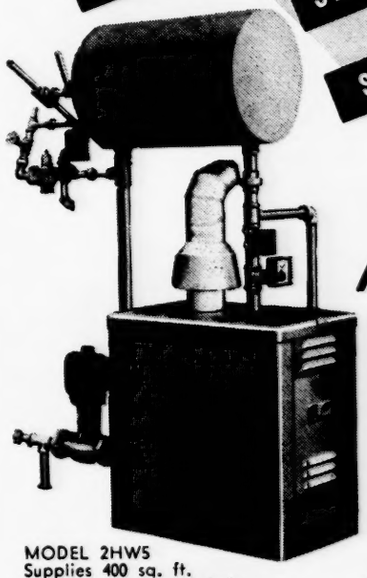


GENERAL G C **CONTROLS**

Manufacturers of Automatic Pressure, Temperature, Level & Flow Controls

Factory Branches: Birmingham (3), Boston (16), Chicago (5), Cleveland (15), Dallas (2), Denver (10), Detroit (8), Glendale (1), Houston (6), Kansas City (2), New York (17), Oklahoma City (1), Philadelphia (40), Pittsburgh (22), Seattle (1), San Francisco (7), St. Louis (12) • **DISTRIBUTORS IN PRINCIPAL CITIES**

SAVES MONEY
SAVES SPACE
SAVES FUEL



MODEL 2HW5
Supplies 400 sq. ft. radiation and 3 gal./min. hot water. Floor space 18" x 27".

New
HYDROTHERM

Domestic HEAT & HOT WATER COMBINATION

- **AGA approved for all gases**
- **Long-life cast iron construction**
- **Numerous patented advantages**

HOOK & ACKERMAN, INC.
18 E. 41st Street
New York 17, N. Y.

Ideal for Low Cost Housing, Garden Apts. Radiant and Convector Heating.

Illustrated Bulletin #1003 on Request.

Infra Insulation, Inc., Dept. HV
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"Simplified Physics of Thermal Insulation."
AN AUTHORITATIVE, 32-PAGE BOOKLET

THE author, Alexander Schwartz, adds to his own findings those of an impressive list of experts and laboratories. The subject headings include: Heat Transfer; Conduction and Density; Convection; Radiation and Emissivity; Rejection, Reflection and Absorption. Other topics are Vapor, Vapor Barriers, Humidity, and Condensation. Every kind of material, mass fibrous insulations and reflective types, how and where to use them, is discussed. There is an enlightening exposition of thermal factors, what they mean, and how different substances compare.

THE famous "Chart of Thermal Insulation Values," a feature popular in previous editions, has been revised and amplified. It contains specially compiled information on *k*, C, R and U factors of all insulations, of all thicknesses, their densities, weights, cubic contents, etc., nowhere else grouped in so convenient form.

INFRA C FACTORS & ROCKWOOL EQUIVALENTS

C.052 Heat Flow Down, equals 6" Rockwool.
C.083 Heat Flow Up, equals 4" Rockwool.
C.10 Lateral Heat, equals 3-1/3" Rockwool.

Thermal Factors Printed on Every Infra Carton

ACCORDION MULTIPLE ALUMINUM & TRIANGULAR REFLECTIVE AIR CELLS

Infra INSULATION, INC.
10 Murray St., N. Y., N. Y.

NEW CATALOGS

Liquid Level Controllers

In Bulletin No. 48-1, both direct-connected and air actuated liquid level controllers are illustrated and described with details of the special construction features and typical applications. Schematic drawings, diagrams, and photographs are included, with capacity charts, flow characteristics, pressure-temperature rating charts, and dimensions. Featured also are Belfield pressure regulating and reducing valve disc-plugs, compression lubricator, stuffing box, angle valves and other auxiliaries.—*Minneapolis-Honeywell Regulator Co., Belfield Valve Div., Philadelphia 44, Pa.*

Item 157

Hot Water Heat

Catalog form CC-549 is devoted to a complete story on the forced circulating Thrush Flow Control System of hot water heat. It contains illustrations, capacity tables, performance charts, and complete catalog copy of all Thrush products. — *H. A. Thrush & Company, Peru, Ind.*

Item 158

Gas Combustion Equipment

Condensed Catalog 292 describes and pictures thirty-three items of gas combustion equipment. Capacity and dimensional tables are included for combustion assemblies, gas-air and air-gas mixers, blowers and boosters, many burner types, pilot and ignition devices, controls, valves, regulators and other accessories.—*Bryant Industrial Div. (Affiliated Gas Equipment, Inc.), 1020 London Rd., Cleveland 10, Ohio.*

Item 159

Crane Cab Coolers

Bulletin 1300, describing air conditioning systems for crane cabs operating in hot or contaminated atmospheres, sets forth in detail the various requirements for air conditioning crane cabs and explains the mechanical and functional features of Dravo's crane cab cooling equipment. Specification data sheets are included for three different models.—*Dravo Corporation, Pittsburgh 22, Pa.*

Item 160

Vapor Equalizing Valves

Bulletin No. 117 describes and illustrates Gorton vapor equalizing valves and air eliminators and includes an installation data sheet. — *Gorton Heating Corp., Cranford, N. J.*

Item 161

Vacuum Heating Pumps

Single and duplex models of electric and steam turbine power vacuum heating pumps are described and illustrated in an 8-page bulletin with dimension and weight charts, capacity charts, and engineering data.—*Whittington Pump & Engineering Corp., 1126 Prospect St., Indianapolis 3, Ind.*

Item 162

LEE DIRECT WARM AIR HEATING

Higher Efficiency! Greater Economy!

Direct warm air heating is fundamentally the most efficient means of transferring heat from fuel to air, and is especially suited to the heating of industrial buildings of all types, hangars, etc. Lee Engineering Company was one of the pioneers in this field, and today offers a complete line of both central system heaters and unit heaters. The Lee system gives heat WHEN needed and WHERE needed. It begins to distribute heat immediately after fire is lighted and fan started, responds promptly without lag. Air-flow is counter-current to flow of combustion gases, so that air becomes warmer as it extracts heat from the progressively hotter gases. Lower installation cost than steam or hot water; requires no licensed attendant; provides ventilation without additional cost. No expensive installation or maintenance of pipes, traps, valves, and pumps. Used with oil or gas, also with coal (stoker or hand-fired). Four main types illustrated here—send for Bulletin 10 with complete facts.

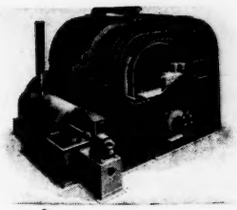
BRICK-SET TUBULAR HEATER

For use with central heating system (with duct distribution). Single heater capacities, 2,800,000 to 8,000,000 Btu per hour. Battery of two operating as unit, provide over 10,000,000 Btu per hour.

Brick Set Tubular Heater

STEEL ENCASED TUBULAR HEATER

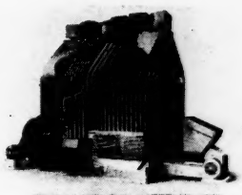
For use with central heating system (with duct distribution). Capacity range, 2,000,000 to 6,000,000 Btu per hour. May be installed in heated area without enclosure, requires no foundation. Unit may be moved by taking apart and reassembling.



Steel Encased Tubular Heater

TUBULAR UNIT HEATER

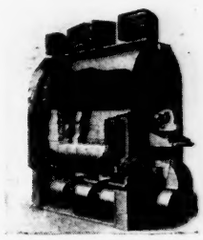
For use as central system (with duct distribution) or as unit heater, with adjustable outlet nozzles. Capacity range, 2,000,000 to 6,000,000 Btu per hour. In sizes up to 4,000,000 Btu, shipped completely assembled with all but mechanical equipment, refractory lining and controls in place. Requires no foundation. Equipped with crane hooks for moving.



Tubular Unit Heater

SHELL UNIT HEATER

For use with or without distributing duct system. Capacity range, 300,000 to 2,000,000 Btu per hour. Both hand and stoker-fired models. All units shipped completely assembled, wired, ready for operation. Units furnished with either refractiory lined or stainless steel combustion chambers. Heaters may be floor mounted or suspended in any position.



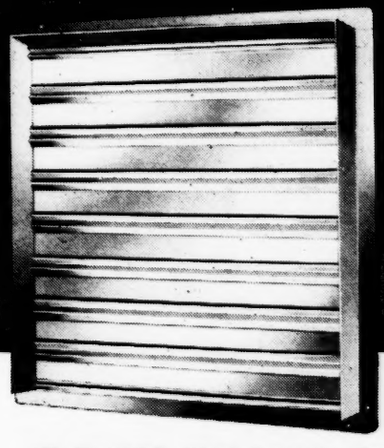
Sheli Unit Heater

LEE ENGINEERING COMPANY

95 RIVER STREET • HOBOKEN, NEW JERSEY

(Formerly Youngstown, Ohio)

they all want **AIR-FLO**



MODEL A-555

THE AUTOMATIC SHUTTER WITH ALL THE FEATURES

Front View—Closed

IT TAKES THE LOAD OFF THE FAN!

Aluminum louvers open fully, permitting capacity fan operation. New heavy reinforcement strip adds strength and long life to the louvers, assures quiet operation and perfect counterbalance, prevents rattling. Deep shroud protects shutter from high winds. Tie-rod, brackets and bearings inside frame, not exposed to weather. Special finish resists corrosion. Many other features.

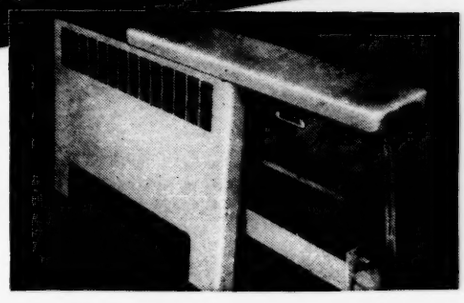
Write for New Air-Flo Catalog 43-F

Illustrations and details of the complete Air-Flo line.

AIR CONDITIONING PRODUCTS CO.

2340 W. LAFAYETTE BLVD. • DETROIT 16, MICH.

LOOK! NEW AMCOIL CONVECTORS



21" high free standing model. Also available in 20" high recessed model. Overall lengths range from 20" to 64".

● Yes, they're *new*, and they're profitable. AMCOIL Convectors have the features people want. Heating elements are all-copper and aluminum, and have the patented Amcoil Fin Surface for maximum BTU capacity. Only AMCOIL Convectors have 2⅛" diameter Copper headers for maximum efficiency. All joints are copper-welded for leak-proof operation. Recessed or free-standing cabinets of heavy steel, baked-enamel finished. Complete with fittings for all steam or hot water heating systems.

You haven't seen your newest chance for convector business until you've seen the new AMCOILS. Send for Bulletin.

Jobbers! Write for information on open territories.

Amcoil

HEATING • AIR CONDITIONING • REFRIGERATION

AMERICAN COILS COMPANY • 360 THOMAS ST. • [illegible]

McCORD
HEATING AND
AIR CONDITIONING
PRODUCTS
BLAST HEATING COILS—AIR CONDITIONING COILS—ALL TYPES
All copper spiral fin tube construction with the tubes formed or bent permits uniform expansion without strain. Headers are heavy wall construction with tubes brazed into extended ferrules formed in the headers. Fins are helically wound and metallically bonded. May we send catalog?
VERTICAL TYPE UNIT HEATERS
A proven heating unit, modern in design, incorporating every unit heater advancement. McCord copper spiral fin tube construction creates air turbulence without undue restriction. Improved fan blades deliver large volume of air with minimum air noise. Guaranteed for 150 lbs. saturated steam pressure. Specify McCord.
HORIZONTAL TYPE UNIT HEATERS
A time proven heating unit, modern streamlined in design, quiet, yet its rugged construction adapts it to all types of installations, guaranteed for use on No. 150 p. s. i. steam pressure. All copper heating element, individual spiral fin tubes free to expand without strain on joints. Standard motors with standard bases used. Wide range of capacities.
Save fuel with McCord Unit Heaters.
McCord CORPORATION
DETROIT 11, MICH.

Full-View Rotameters

A 4-page bulletin No. 6, describes the full-view rotameter with safety shielding which features low maintenance, corrosion resistance, operating safety, and high accuracy. Five new additions to the line are also included.—*Brooks Rotometer Company, P. O. Box No. B-63549, Lansdale, Pa.*

Item 163

Shuntflo Motors

Bulletins 400-F1 and 400-F2 describe and illustrate, respectively, models SMKS and SMDH shuntflo meters for steam, air or gas flow measurement. Installation diagrams, capacity charts, typical specifications, and capacity tables are included. Model SMDH has an integrating counter for totalizing flow and is installed directly in 1-inch or 1½-inch pipelines. Model SMKS measures flow in lines from 1 to 12 inches in diameter. —*Builders-Providence, Providence 1, R. I.*

Item 164

Metal Frame

Unistrut metal framing, a patented method for building frames, hangers and supports for electrical, plumbing, heating and air conditioning equipment, is described in Catalog 500, 2 colors, 24 pages. Special forms and accessories are said to eliminate drilling and welding and result in a completely adjustable, reusable frame structure.—*Unistrut Products Company, 1013 West Washington Blvd., Chicago 7, Ill.*

Item 165

Hospital Control Systems

A new booklet describes specialized heating and air conditioning control requirements for hospitals and sets forth in non-technical terms the various atmospheric requirements of different hospital rooms and areas. Also discussed are a number of fuel-saving heating control systems which may be installed in existing buildings as well as those under construction. —*Minneapolis-Honeywell Regulator Co., Minneapolis, Minn.*

Item 166

Refrigeration Computation Charts

Charts, formulas and examples for figuring number and size of quick-action or hold-over plate type evaporators are included in the new Kold-Hold catalog. Other data include specific and latent heats of perishable products, infiltration loads, insulation values, etc. Catalog is indexed and sectionalized. — *Kold-Hold Manufacturing Co., Lansing, Mich.*

Item 167

Advertising Aids

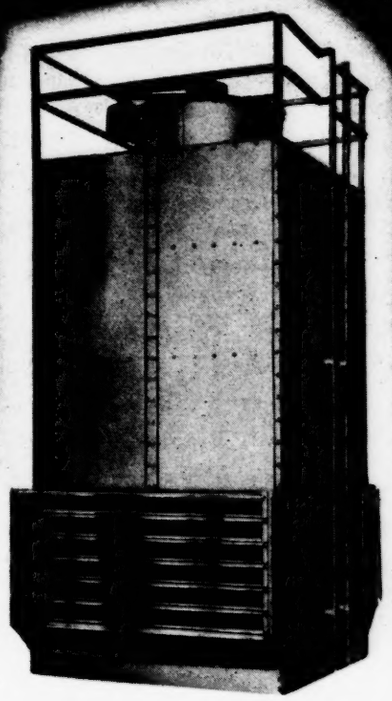
A supplement to *The Publishers' Auxiliary* consists of a dealer ad-help reporting service which lists the various advertising mats and cuts available from manufacturers in various fields, including 79 companies in the heating, plumbing, and ventilating fields. — *The Publishers' Auxiliary, Dealer Ad-Help Editor, Thorn Hill, Frankfort, Ky.*

Item 168

NEW MARLEY *Verflow**

An "*Extra Quality*" Tower

. . . At No Extra Cost



The New Marley VERFLOW sets a new standard in air conditioning and refrigeration Cooling Towers.

Check these "extra quality" features formerly found only on heavy duty, industrial cooling towers that are now available on all Marley wood or steel VERFLOWS at no extra cost to you . . .

√ Lifetime nail-less filling. √ Filling that can be removed and reinstalled in a matter of minutes. √ Totally enclosed motors. √ Enclosed bearing housings with grease packed bearing. √ Marley air-foil section cast aluminum alloy fans. √ Special corrosion-resistant paint on metal towers. √ Heavy double-sheathed wood casing on wood towers. √ Patented, close-centered Marley jet action spray nozzles in balanced distribution system. √ Hinged louver frames and removable louvers. √ Quickly erected, easily operated with little maintenance time.

These and other "extra quality" features make Marley VERFLOW the best buy on the market. Many special accessories such as Geareducer drive, handrail and ladder are also available at a small added cost.

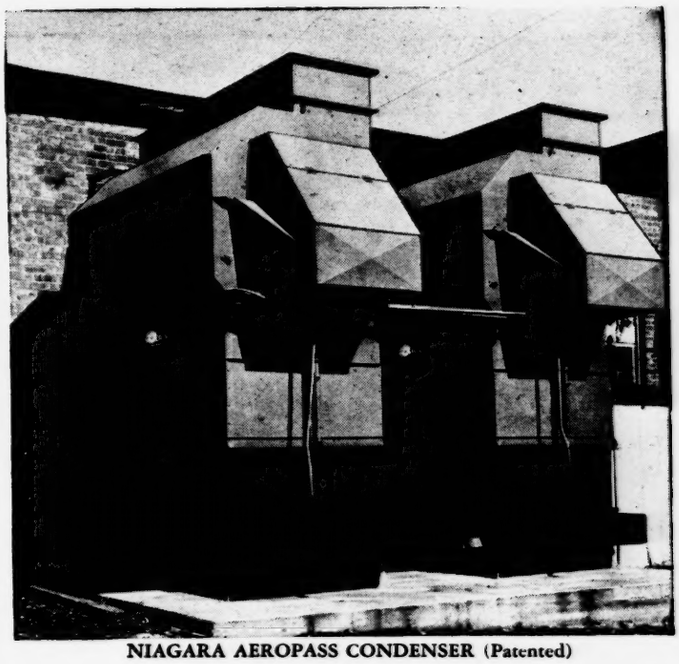
Ask your nearest Marley representative to tell you more about the VERFLOW or write for Bulletin V-49.

MARLEY COOLING TOWERS

*VERFLOW is the exclusive trademark of

THE MARLEY COMPANY, INC.

Kansas City 15, Kansas



NIAGARA AEROPASS CONDENSER (Patented)

Saves half the Engineer's Troubles ...with a Refrigeration Plant

Hundreds of engineers have been interviewed about their experience with the Niagara Aeropass Condenser. In one way or another, they say, "I wouldn't go back to anything else." And their managers, who watch the costs, say, "Best investment we ever made, couldn't operate now, without it."

Niagara Aeropass Condensers have three exclusive features which save trouble and money in running a refrigeration plant:

The "Duo-Pass"— keeps scale and salts from crusting the coils, keeps the condenser always at full capacity.

The "Oilout"— removes oil and dirt from the refrigerant, at the exact point where the oil vapor is condensed and the refrigerant is not.

The "Balanced-Wet-Bulb" control gives automatic operation at the minimum head pressure, saving power cost the year 'round!

In addition, the Niagara Aeropass Condenser saves nearly all your cost of cooling water, quickly bringing back to you the cost of installation. In plants where refrigeration is a production process, owners know that this condenser has reduced their costs.

Write for Bulletin 103.

You can see one of these installations near you.

NIAGARA BLOWER COMPANY

Over 35 Years of Service in Industrial Air Engineering

Dept. HV, 405 Lexington Ave., New York 17, N. Y.

District Engineers in Principal Cities

INDUSTRIAL COOLING HEATING • DRYING

NIAGARA

HUMIDIFYING • AIR ENGINEERING EQUIPMENT

ANOTHER

New York City Housing Project Installs

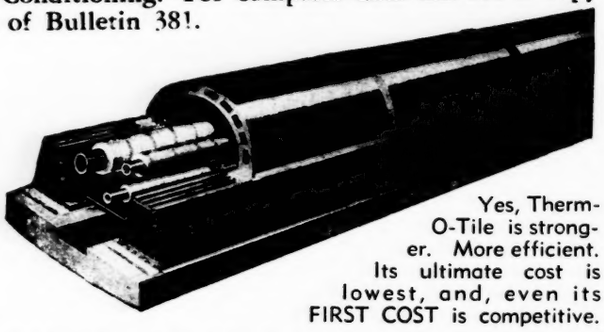
THERM-O-TILE

Reg. U. S. Pat. Off.

Underground Steam Conduit . . .

PELHAM PARKWAY HOUSES

are the latest. In previous ads we named nine of these huge New York City projects that have selected Therm-O-Tile. See, for example, our ad in the January issue of Heating, Piping and Air Conditioning. For complete data ask for a copy of Bulletin 381.



Yes, Therm-O-Tile is stronger. More efficient. Its ultimate cost is lowest, and, even its FIRST COST is competitive.

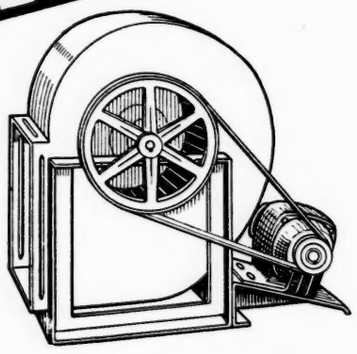
Sold and installed by Johns-Manville Construction Units in all Principal Cities.

H. W. PORTER & CO., Inc.

823-V Frelinghuysen Ave. Newark 5, N. J.

HY-DUTY BLOWERS

● There's a Hy-Duty properly sized for hundreds of jobs where large volume, low velocity and quiet are demanded—double inlet or single—top or bottom motor mounting—all discharge positions—200 c.f.m. to 15,000—heavily constructed—looks and finish to stack up with the best jobs and priced right.



Announcing . . .

A new, high powered, quiet, direct driven, single inlet Hy-Duty blower with a wheel 5" diameter, 3" wide. Immediate delivery, attractive price, suitable for a multitude of small air handling jobs.

AIR HANDLING AT ITS QUIETEST BEST

● SEND FOR LITERATURE AND ENGINEERING INFORMATION

BLOWERS • ATTIC FANS • EXHAUST • PORTABLE OR WINDOW FANS

SCHWITZER-CUMMINS COMPANY

VENTILATING DIVISION • INDIANAPOLIS 7, INDIANA

Fine Blowers and Fans for over 30 Years

Direct Fired Heaters

Use of direct fired heaters in processes, spray and tunnel drying, or pre-heating, etc., are described and illustrated with diagrams in color in a 6-page, 4-color bulletin No. 600.—*Peabody Engineering Corp., 580 Fifth Ave., New York 19, N. Y.*

Item 169

Low Pressure Boilers

The National line of low pressure commercial steel boilers, manufactured in 16 sizes with ratings from 2,500 to 35,000 square feet of steam radiation are described and illustrated in a 6-page bulletin, Catalog 507, with specifications, ratings, and dimensions.—*The National Radiator Co., Johnstown, Pa.*

Item 170

Exhaust Fans

Emerson Electric exhaust fans for business, industrial and institutional buildings are described and illustrated in a 16-page Catalog No. X6259. Line includes direct and belt driven fans and window ventilating fans. Catalog gives design and construction details, specifications, and performance data.—*The Emerson Electric Manufacturing Co., St. Louis 21, Mo.*

Item 171

Fans in Power Houses

Applications of fans, blowers and dust collectors in power plants, including data on the Gyrol fluid drive, are shown in a spiral bound booklet "Progress in Power," consisting of a series of advertisements by the manufacturer.—*American Blower Corp., 6000 Russell St., Detroit, Mich.*

Item 172

Regulating Valves

Regulating valves for water, steam, air, gas or oil are described and illustrated with schematic diagrams, dimension tables and operating data in Bulletin No. 10-1.—*Minneapolis-Honeywell Regulator Co., Belfield Valve Div., Philadelphia 44, Pa.*

Item 173

AUGUST, 1949

TO OBTAIN COPIES OF TRADE LITERATURE listed in this issue, circle on the list below the publication wanted, using item number at the end of each review. Print your name and address, mail to

Editor, HEATING AND VENTILATING
148 Lafayette Street, New York 13, N. Y.

157	158	159	160	161	162
163	164	165	166	167	168
169	170	171	172	173	

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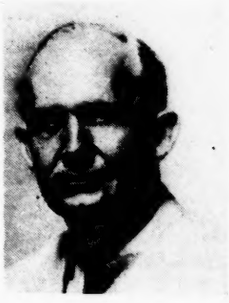
Getting Personal

R. J. Lorenzi (co-author, *Effect of Thermostat Characteristics on Performance of a Panel Heating System*, page 69) has specialized for the past twelve years in heat transfer with emphasis on design and research. He attended Wesleyan University and the Yale Graduate School, and from 1938 to 1943 was associated with a professional engineering firm specializing in heating and air conditioning design for commercial and industrial buildings. In his early work with the Pierce Foundation during the war, his experience included the planning and execution of thermal conductivity tests to evaluate the performance of military clothing.

In his present capacity as research engineer for the Pierce Foundation, Mr. Lorenzi's work has brought him into intimate contact with panel heating, particularly with systems of the electrical type.

He is a member of the American Society of Heating and Ventilating Engineers, and is co-author of three papers which have been published in the transactions of this society.



L. P. Herrington

R. J. Lorenzi

Dr. L. P. Herrington (co-author, *Effect of Thermostat Characteristics on Performance of a Panel Heating System*, page 69) is director of research at the John B. Pierce Laboratory of Hygiene in New Haven, and Research Associate Professor in Public Health at Yale University. He is a graduate of Stanford University, and has held faculty appointments at the University of Illinois and Yale, and a Rockefeller Fellowship at the Charite Hospital in Berlin.

Dr. Herrington's primary field of interest is environmental physiology and psycho'ogy, which is concerned with the effects of heat, light, sound, and similar factors on human health and efficiency. As a member of the New Haven Laboratory of the Pierce Foundation, he has conducted and published a long series of research studies centering on thermal physiology with particular reference to the problems of hygienic air conditioning, heating and ventilating. Many of these studies have been summarized in a recent book *(Temperature and Human Life* Princeton U. Press, 1949) co-authored with C.-E. A. Winslow.

. . . Since the Last Issue

Dr. Louis C. McCabe, former coal branch chief in the **Bureau of Mines** and for the last two years director of the Los Angeles, Calif., Air Pollution Control District, has rejoined the Federal Bureau as chief of air and stream pollution prevention research with headquarters in Washington, D. C.

Clisby Associates, Inc., 904 Sixth Ave., N. Birmingham, Ala., has been appointed distributor for Dravo Counterflo heaters in central and northern Alabama, by **Dravo Corp.,** Pittsburgh, Pa. The company also will distribute Dravo Crane Cab Coolers in all of Alabama and northwest Florida. The Alabama heater territory does not include DeKalb, Madison or Jackson Counties.

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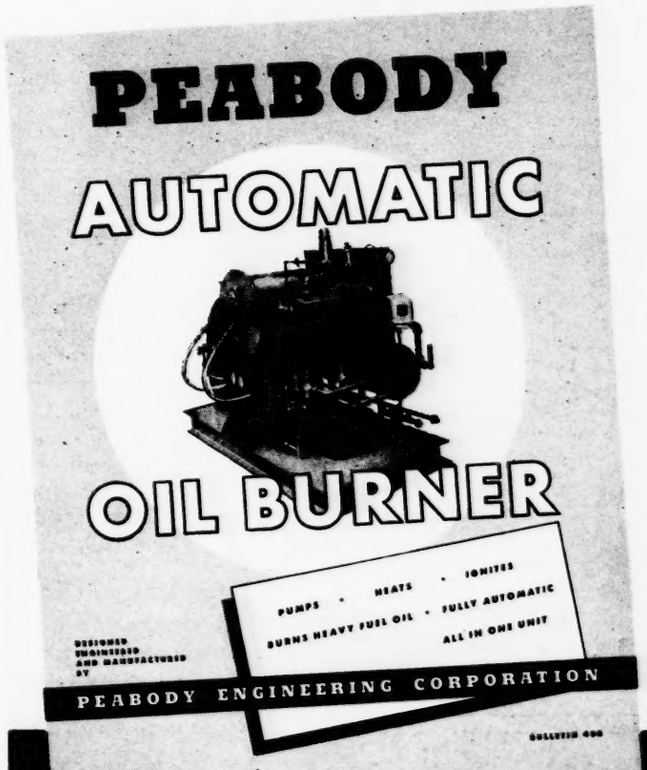

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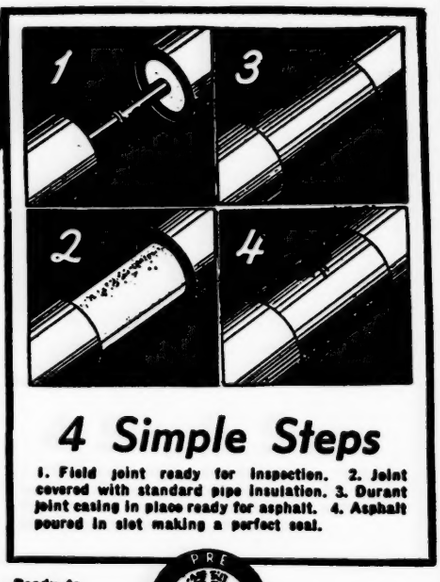

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The following men have recently joined the **Kennard** field organization: **Chet Adams Co.**, Greensboro, N. C.; **John J. Barry**, Baltimore, Md.; **M. Blazer & Son**, Passaic, N. J.; **John R. Ellstrom**, Philadelphia, Pa.; **Don E. Hawk Associates**, New York, N. Y.; **Paller Engineering Co.**, Indianapolis, Ind.; **J. Fuller Stafford** and **C. L. Benson Associates**, St. Paul, Minn.; and **R. F. Zimmerman & Co.**, Shreveport, La. These men are now available, in the territories listed, for information on application of Kennard's complete line of heat transfer equipment.

R. H. Warmee

Roy H. Warmee, formerly sales manager Moduflow division and prior to that manager sales promotion division of Minneapolis-Honeywell Regulator Co., has joined the **White Manufacturing Co.**, 2362 University Ave., St. Paul, as vice president and sales manager. Before going with Minneapolis-Honeywell, Mr. Warmee spent ten years with the Philadelphia Coke Co. of Philadelphia, a Koppers Co. subsidiary, merchandising coke, heat regulators and coke stokers. Before that he was with the Koppers Company of Pittsburgh engaged in the construction of coke and gas plants, among which were Consolidated Edison of New York and the Brooklyn Union Plant at Brooklyn. While Mr. Warmee was with the Philadelphia Coke Co., they won the Howard G. Ford national award for outstanding achievement in sales management. In 1947 he was awarded the Edmund F. Maier trophy for meritorious service to sales management and civic leadership in Minneapolis.

The appointment of **Franklin A. Ulrich** as controller of **Fedders-Quigan Corp.** has been announced. Mr. Ulrich was a partner and senior auditor of the accounting firm of Meech, Harmon, Lytle and Blackmore, from which he has resigned to accept his new position.

David L. Fiske, consulting engineer, 111 Broadway, New York, N. Y., has been engaged by the **Economic Cooperation Administration** for a three-month survey of refrigeration in the Mediterranean area, where a number of freezing, storage and food processing plants are projected.

J. F. Wulfetange, Jr., 317 State St., Syracuse 2, N. Y., has been appointed eastern New York representative for the **Automatic Control Co.** of St. Paul, Minn., manufacturers of automatic liquid level controls. Mr. Wulfetange was graduated from Penn State with a BSEE degree. He has had recent association with manufacturers of high quality instruments.

Ralph Hooke, middle west regional manager for the **Timken Silent Automatic Division,** Jackson, Mich., died in an Evanston, Illinois, hospital June 8, after a short illness. Joining Timken Silent Automatic as a district sales manager in 1931, Mr. Hooke took over as middle west regional manager in 1937.

The **Wolverine Tube Division of Calumet and Hecla Consolidated Copper Co.** has announced the opening of a new Southeastern District Sales Office at 788 Spring St., N.W., Atlanta, Ga. **A. R. Kaspark** will cover north and central Georgia, eastern Tennessee and western counties of North and South Carolina from this office. M. J. Cook and T. P. Stone will continue to represent the company in their respective territories—the western and southern sections of the southeast area.

Professor William H. Scheick, coordinator of the Small Homes Council at the University of Illinois, has been granted a year's leave beginning September 1 to serve as the first executive director of the newly organized building research Advisory Board in Washington, D. C. The board was organized by the National Research Council and operates under its jurisdiction.

William A. Mulcock has been appointed manager of sales administration and forecasting of the automatic heating division of the **General Electric Co.'s** air conditioning department.

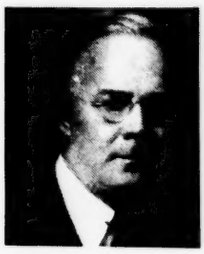
Jan-Air, Inc. of Richmond, Ill., has recently purchased the fan wheel division of the **Janette Manufacturing Co.** of Chicago, specializing in fans for the oil burner, coal stoker, air conditioning, and in industrial cooling varieties. **John I. Janette,** formerly vice president and plant manager of the Janette Manufacturing Co., heads up this new organization.

Robert J. Mullally, district sales manager in charge of the Detroit office for **The Youngstown Sheet and Tube Co.,** has retired and has been succeeded by **C. Hix Jones,** formerly assistant to Mullally. **D. L. Markle, Jr.** has been appointed district sales manager in charge of the New Orleans office for Youngstown, succeeding **Orville B. Ewing** who died recently.

S. B. Withington, general manager of the Lycoming and Spencer Heater plants, has been elected a vice president of **Avco Manufacturing Corp.**

H. Dale Cook has been named manager of the industrial control sales division of the **Perfex Corp.,** Milwaukee manufacturers of automatic heating controls. Mr. Cook succeeds A. B. Meeg who recently left Perfex to join Bell & Gossett.

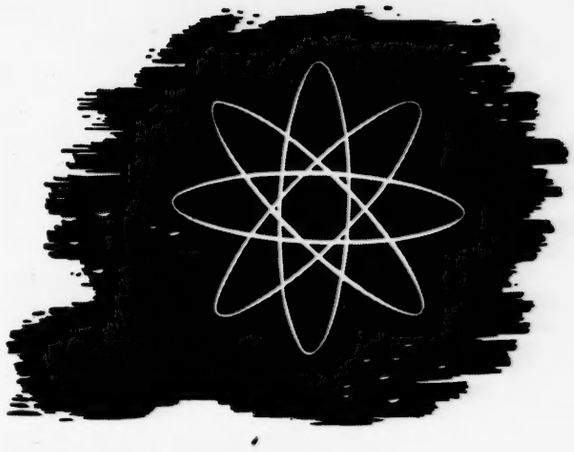
Robert D. Welsh, formerly assistant sales promotion manager of the Thermoid Co., has been appointed advertising manager of the **W. B. Connor Engineering Corp.,** manufacturers of activated carbon air purification equipment and ceiling air diffusers.



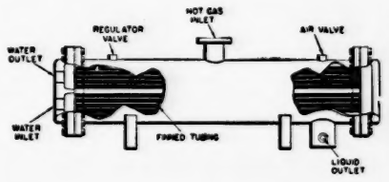
J. D. Berg

J. D. Berg, chief executive officer and a director of **Dravo Corp.,** Pittsburgh, passed away suddenly, Wednesday night, June 29, in his home in Chatham, Cape Cod, Mass., at the age of 66. Mr. Berg had headed Dravo Corporation since the death of its founders F. R. and R. M. Dravo in 1934. He served as board chairman from 1934 to 1946, at which time he was elected to his present position. Although Mr. Berg resided for the past two years in Chatham, Mass., he remained active in the affairs of Dravo Corporation. Mr. Berg became identified with Dravo-Doyle Co., now a subsidiary of Dravo Corp., in 1906 as sales engineer, becoming vice president in 1908 and president in 1923. He was prominent in civic affairs as well as being an outstanding Pittsburgh industrialist. He was a director of Dravo-Doyle Co., Ful'erton-Portsmouth Bridge Co. and Consolidated Natural Gas Co. Mr. Berg graduated from Lehigh University in 1905 and had served as a trustee of that institution for a number of years. His degrees include that of M.E. in 1905 and Doctor of Engineering (honorary) in 1947.

C. A. Dunham Co. announces the following July changes in its sales organization: Sales engineer **W. C. Zingheim** of Chicago retires. The company will operate a Chicago sales and service office to serve the metropolitan area and ten adjoining counties in Illinois. It will be under the management of **George A. Mansinger.** Associated with him are **E. A. Taylor, Carl King** and **Paul Schaeffer.** Sa'es engineer **W. H. Baldwin** of Detroit retires and is succeeded by **C. C. Grubb** formerly at Akron, Ohio. After 37 years as sales engineer, **Wallace Hoeing** of Louisville retires from all business activity. He is succeeded by **David J. Rank** formerly located at Indianapolis. **William W. Evans** succeeds C'air Grubb (removed to Detroit) as sales engineer and continues at the former Grubb address. Sales engineer **R. K. Rouse** of Greenville, S. C. has established an office at Columbia with **W. M. Plowden** as local manager.

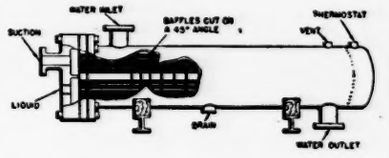


Electronic tube expanding for Patterson *freon coolers and condensers*



Patterson freon condenser
type O fin tube construction

Electronic Engineering has found the way to provide automatic, regulated control when expanding tubes in the tube sheet. The operator sets the electronic control at a predetermined point, selected for the kind of metal used and size of tube to be expanded. Just the right pressure and timing—automatically.



Patterson freon cooler
dry expansion type

Electronic tube expanding eliminates the chances of human error . . . chances of either overexpanding, resulting in overstressed metal or underexpanding, resulting in probabilities of early leakage and high maintenance.

In our shop, all tube expanding operations are carried on under Precision Electronic Control. Metal to metal joints are tight, strong, and safe . . . a plus value to be found in all Patterson-Kelley Freon Coolers, Condensers, Heat Exchangers and other tubular products.

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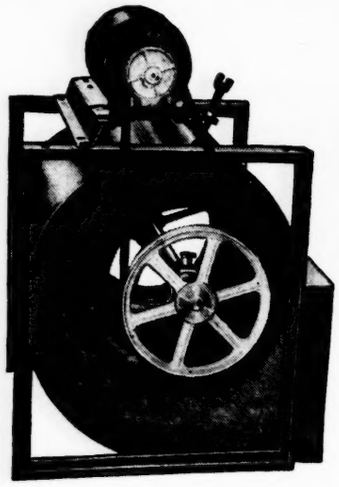

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MASSACHUSETTS . .

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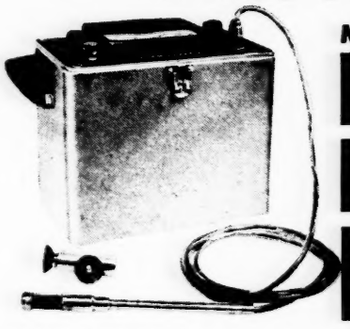

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FROM 0 TO 4 NEG. AND FROM 0 TO 10 POS. IN. WG

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Wayne McCarthy has been promoted to the position of North coast regional manager of the **Minneapolis-Honeywell Regulator Co.** Mr. McCarthy succeeds John B. Banks, who has resigned to take a position with another firm. Mr. McCarthy, a native of Helena, Mont., spent five years in the contracting business at Mitchell, S. D., before joining the Honeywell sales organization in 1937. **Wayne E. Petersen,** who has been on the company sales staff in Des Moines, will succeed McCarthy as Des Moines branch manager.

J. F. Byers, Sr.

J. Frederic Byers, Sr., chairman of the board of **A. M. Byers Co.,** Pittsburgh, died June 11 in Roosevelt Hospital, New York City. Funeral services were held at Sewickley, Pa., June 13, with interment in Allegheny Cemetery, Pittsburgh. He was the son of the late Alexander MacBurney Byers, who founded A. M. Byers Company in 1864. Among the survivors are two sons, J. Frederic Byers, Jr., and Buckley Morris Byers, both of whom are directors of the company. Mr. Byers also was a director of Westinghouse Air Brake Company, Union Switch & Signal Company, Western Allegheny Railroad, and Mellon National Bank and Trust Company of Pittsburgh. He was a trustee of the Carnegie Institute of Techno'ogy and the Carnegie Library, both of Pittsburgh.

Timken Silent Automatic Division announces the appointment of **Willard J. Hatton** as coal burner project engineer. For the past eight years Mr. Hatton served on the technical staff of the Battelle Memorial Institute Laboratories, where he was engaged in the design and development of automatic residential stokers.

The Revere Quality House Institute has become the Revere Quality House Division of the Southwest Research Institute, it has been announced jointly by C. Donald Dallas, chairman of the board, **Revere Copper and Brass Incorporated,** and Dr. Harold Vagtborg, president of **Southwest Research Institute.** The announcement invited further builder participation in the program, which is conducted without cost to the builders themselves. The builder phase of the activities is centralized in a New York office, maintained by the Builder Relations Department, Revere Quality House Division, Southwest Research Institute, at 280 Madison Ave., New York 16, N. Y. C. W. Smith is director of the division, and John Hancock Callender architectural consultant.

C. M. Wilson

C. Milton Wilson has been appointed sales manager of **Anemostat Corp. of America.** Mr. Wilson, formerly manager of sales of Ingersoll Division of Borg-Warner Corp., was also associated for many years with Clyde R. Place, consulting engineer, on major New York building projects. During the war he was a production engineer for the Army Air Forces at Wright Field. He was 'ater transferred to the Manhattan District Project as an executive engineer.

John R. Lenox has been named manager of industrial relations and the personnel department of **Minneapolis-Honeywell Regulator Co. Sherwood M. Sitz,** who has been directing industrial relations at the company's Minneapolis factories, has been promoted to director of industrial re'ations in a consulting capacity for the entire organization, including subsidiary companies. **George B. Benton,** who has been personnel manager, has been transferred to the company's expanded insurance department and will make his headquarters at the home office in Minneapolis.

Professor Mario Carl Giannini of **New York University's** College of Engineering has been appointed assistant dean in charge of the day division. The post of associate professor of mechanical engineering has been held by Mr. Giannini since 1943. In addition, he has been executive assistant to the College of Engineering since 1946, and has carried on his own private consulting practice since 1931. Born in 1902 in New York City, Professor Giannini received his Bachelor of Science degree in Mechanical Engineering from New York University in 1923. He received the degree of Mechanical Engineer in 1926 from the same institution.

Appointment of **Fred H. Pillsbury** as vice president in charge of operations of the **Century Electric Co.** of St. Louis, Mo., has been announced. At the time of his promotion he was serving as executive engineer.

John J. Summersby has been elected vice president in charge of sales by the board of directors of **Worthington Pump and Machinery Corp. Thomas J. Kehane** was made assistant vice president and general sales manager. Mr. Summersby joined the Cincinnati works of Worthington in 1916 as a student engineer. In 1919 he was assigned to the St. Paul district office as a salesman and later was made district sales manager. He was manager of Holyoke works sales from 1929 to 1931; assistant general sales

J. J. Summersby

T. J. Kehane

manager from 1931 to 1934; and since, assistant vice president and general sales manager. Thomas J. Kehane was born in Jersey City. He studied engineering and architecture at Mechanics Institute and Pratt Institute, after joining Worthington in 1915 as a fourteen-year-old office boy. Since then he has advanced through various positions in the sales department, and in 1944 was appointed commercial vice president, Pacific Coast. For the past two years he has served as Pacific Coast regional vice president of the Navy Industrial Association.

Charles V. Fenn, formerly in charge of direct sales for **Carrier Corp.** in the southeastern section of the United States, has been appointed Carrier's manager of direct sales with headquarters in Syracuse. **Russell H. Gray** has been named manager of direct sales in the Atlanta district to replace him. Mr. Gray was assistant manager of direct sales in Syracuse.

F. S. Cornell has been named manager of the water heater division of the **A. O. Smith Corp.** Mr. Cornell had been assistant manager of the division, which operates the Kankakee, Ill., works of the company.

Benjamin T. Hain, vice president for manufacturing, and **Carroll M. Baumgardner,** vice president for sales, have been elected to the board of directors of **The National Radiator Co. John G. dePass,** general credit manager for the company, has been elected assistant secretary. The company's Trenton, N. J., manufacturing plant has been reopened because of recent increases in orders for cast iron radiators. Manufacturing operations at the Trenton plant were suspended in April.

WINDSOR VILLLAGE
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L & L BLDG. CORPORATION
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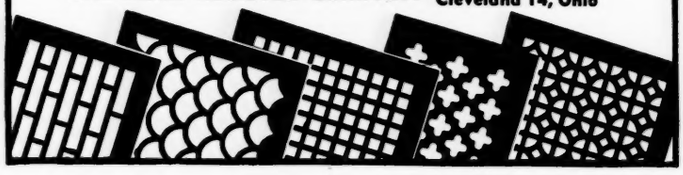

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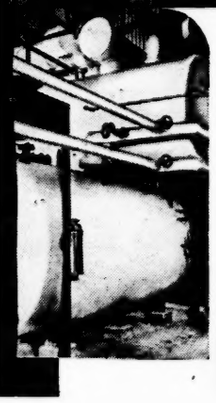

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1835 S. 54th Ave., Chicago 50, Ill.
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Adhesives?... ARABOL!

C. W. Colby of Whitewater, Wisconsin, has been elected executive vice president and a director of **Thermek Corp.** of Chicago. Mr. Colby designed and supervised the installation of the largest single application of oil furnaces in 57 buildings of Mount Holyoke College at Holyoke, Mass. He designed and supervised the installation of the largest single radiant heating system in the world in the Kankakee plant of the A. O. Smith Corp.

During recent years Mr. Colby has been head of heating research for A. O. Smith of Milwaukee in charge of the Whitewater Laboratory. The Laboratory facilities at Whitewater, by arrangement with A. O. Smith Corp., are now available to Thermek and its licensees.

Canadian Degree-Days for May, 1949*

City	May		Cumulative, Sept. 1 to May 31	
	1949†	Normal	1948-49	Normal
Calgary, Alta.	375	480	9358	9111
Charlottetown, P.E.I.	521	536	7700	8263
Crescent Valley, B. C.	291	403	8109	7731
Edmonton, Alta.	375	428	10,016	9797
Fort William, Ont.	552	567	9550	10,045
Grand Prairie, Alta.	499	505	10,931	10,122
Halifax, N. S.	484	493	6710	7395
London, Ont.	252	307	6703	7275
Medicine Hat, Alta.	243	301	8835	8495
Moncton, N. B.	465	474	7907	8568
Montreal, P. Q.	309	325	7299	8413
North Bay, Ont.	471	434	8842	9180
Ottawa, Ont.	330	310	7720	8674
Penticton, B. C.	202	273	7125	6346
Porqu.s Junction, Ont.	598	558	10,152	11,144
Prince George, B. C.	477	490	9817	8996
Quebec City, P. Q.	347	428	7823	9276
Regina, Sask.	332	434	10,592	10,891
St. John, N. B.	499	505	7294	8081
Saskatoon, Sask.	346	397	10,694	10,616
Toronto, Ont.	243	341	6116	7236
Vancouver, B. C.	301	326	5897	5303
Victoria, B. C.	301	366	5254	4935
Windsor, Ont.	205	251	5886	6705
Winnipeg, Man.	370	397	10,326	10,841

*These data are supplied through the courtesy of the Meteorological Division, Air Service Branch, Department of Transport, Canada.

†These figures were not available at the time last month's issue went to press.

COMING EVENTS

Where listed, names or titles of individuals are those from whom further information is available.

AUTOMOTIVE ENGINEERS MEETING—S A E National West Coast Meeting, at Multnomah Hotel, Portland, Ore. John A. C. Warner, secretary, Society of Automotive Engineers, 29 W. 39th St., New York 18, N. Y.....**AUGUST 15-17, 1949.**

INSTRUMENT CONVENTION—Fourth National Instrument conference and exhibit of the Instrument Society of America, at Municipal Auditorium, St. Louis, Mo. John McCaffery, assistant executive secretary of Society, 1117 Wolfendale St., Pittsburgh 12, Pa.....**SEPTEMBER 12-16, 1949.**

PLUMBING AND HEATING MEETING—Annual meeting of the Plumbing and Heating Industries Bureau, at Palmer House, Chicago, Ill. Norman J. Radder, secretary of Bureau, 35 E. Wacker Dr., Chicago 1, Ill. .....**OCTOBER 5, 1949.**

WELDING AND METALS MEETING—1949 annual meeting of the American Welding Society, in conjunction with the National Metals Congress, tentatively at Cleveland Hotel, Cleveland, Ohio. Secretary of Society, 33 W. 39th St., New York 18, N. Y. .....**OCTOBER 16, 1949.**

AGA CONVENTION—1949 annual convention, American Gas Association, at Chicago, Ill. For details, contact secretary, AGA, 420 Lexington Ave., New York 17, N. Y. .....**OCTOBER 17-20, 1949.**

PUBLIC HEALTH MEETING—77th annual meeting of the American Public Health Association, and meetings of related organizations, at the Hotels Statler and New Yorker, New York City. Dr. Reginald M. Atwater, executive secretary of the Association, 1790 Broadway, New York, N. Y. .....**OCTOBER 24-28, 1949.**

ENGINEERING AND POWER EXPOSITION—Ninth midwest and power exposition, at Navy Pier, Chicago, Ill. George E. Pfisterer, 308 W. Washington St., Chicago, managing director. .....**NOVEMBER 11-16, 1949.**

REFRIGERATION SERVICE CONVENTION—12th annual convention, Refrigeration Service Engineers Society, at The Ambassador and Ritz Carlton hotels, Atlantic City, N. J., in conjunction with 6th All-Industry Refrigeration and Air Conditioning Exposition. W. Vernon Brumbaugh, executive secretary of REMA, 1346 Connecticut Ave., N.W., Washington 6, D. C. .....**NOVEMBER 13-16, 1949.**

REFRIGERATION SHOW—6th All-Industry Refrigeration and Air Conditioning Exposition, at Atlantic City, New Jersey. R. K. Hanson, show director, REMA, 1107 Clark Building, Pittsburgh 22, Pennsylvania. .....**NOVEMBER 14-18, 1949.**

INDUSTRIAL HYGIENE MEETING—14th annual meeting and professional conferences of Industrial Hygiene Foundation, at Mellon Institute, Pittsburgh, Pa. Secretary of Foundation, 4400 Fifth Ave., Pittsburgh, Pa. .....**NOVEMBER 16-18, 1949.**

CHEMICAL EXPOSITION—22nd exposition of Chemical Industries, at Grand Central Palace, New York. Charles F. Roth, manager of the International Exposition Company, Grand Central Palace, New York 17, N. Y. .....**NOVEMBER 28—DECEMBER 3, 1949.**

ASRE CONVENTION—46th annual convention, American Society of Refrigerating Engineers, at Edgewater Beach Hotel, Chicago, Ill. M. C. Turpin, secretary of Societv. 40 W. 40th St., New York 18, N. Y. .....**DECEMBER 4-7, 1949.**

PLANT MAINTENANCE SHOW AND CONFERENCE — First plant maintenance show and conference, in the Auditorium, Cleveland, Ohio. Clapp & Poliak, Inc., 350 Fifth Ave., New York 1, N. Y., exposition management...**JANUARY 16-19, 1950.**

HEATING AND VENTILATING MEETING—56th annual meeting of the American Society of Heating and Ventilating Engineers, at Dallas, Tex. A. V. Hutchinson, secretary, 51 Madison Ave., New York 10, N. Y.....**JANUARY 23 26, 1950.**

HEATING AND VENTILATING SHOW—Southwest Air Conditioning Exposition of the International Heating and Ventilating Exposition, State Fair Grounds, Dallas, Texas, under the auspices of the American Society of Heating and Ventilating Engineers. Charles F. Roth, manager of the International Exposition Company, Grand Central Palace, New York 17, N. Y. .....**JANUARY 23-27, 1950.**

CONTRACTORS MEETING—1950 annual meeting of the Heating, Piping, and Air Conditioning Contractors National Association at St. Francis Hotel, San Francisco, Calif. Secretary, of Association, 1250 Avenue of the Americas, New York 20, N. Y. .....**MAY 8-12, 1950.**

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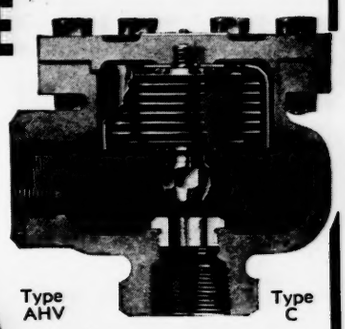

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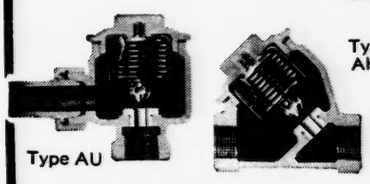

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